NAVAL PROPULSION GEARS

PROGRESS AND DEVELOPMENT 1946—1962

BY

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INTRODUCTION

It was in 1910 that Sir Charles Parsons installed reduction gearing in S.S. *Vespasian* and this was followed by the large scale introduction of geared turbine machinery into the Fleet of the First World War period. Tostevin[1] has described the experience with the gears of these ships :

'. . . of the 596 sets of all-gear installations on service in the Navy, some extending up to nearly six years, it has only been necessary to remove three for refit . . . and here it must be emphasized that no actual breakdown occurred and the gears, after dressing up and, in one case, new pinions being supplied, were subsequently re-utilized '.

This was a remarkable record of achievement and a striking tribute to the gear engineers of those days.

The experience with turbine reduction gears during the Second World War was less praiseworthy and Joughin[2] has described some of the failures which occurred. His paper invites the conclusion that the comparatively small increases in gear loading since the early days had brought with them a disproportionate sacrifice of reliability. Furthermore, the ships of the Royal Navy were fitted with single-reduction gears whereas, in the U.S.A., double-reduction gears had been successfully developed before the war and were fitted in a very large number of U.S.N. warships.

In 1946, with recent war experience in mind, the Admiralty-Vickers Gearing Research Association (A.V.G.R.A.) was formed as an association between the Admiralty and certain industrial firms, representing the user, gear manufacturer and gear-cutting machine maker. In due course, B.S.R.A. (ex Pametrada) became a member of the Association which, since its formation, has been greatly assisted by the staff and facilities of the National Engineering Laboratory.

A.V.G.R.A.'s objectives were, briefly, improvement in gear cutting accuracy, the development of post-hobbing processes, evaluation of alternative materials and the development of surface hardened and ground gears. Much of the work carried out has already been described by Braddyll[3], Chamberlain[4], Chesters[5], Newman[6] and Page[7] and this work forms the background of Part I of the present paper in which a number of post-war naval gear designs are described and an account is given of their performance on trials and in service.

These designs are presented in chronological order and it will be seen how each has been influenced by the advances in knowledge and manufacturing techniques which have resulted from A.V.G.R.A.'s work.

E.P. lubricating oils are used in the majority of ships designed and built since the war and in Part II of the paper, experience with these oils is described and the cleaning and flushing of gearing is discussed. An account of experience with prefinished steel backed bearings is also given. Although carburized gears have been used extensively in recent designs, the distortion which occurs during hardening necessitates excessive grinding and thus increases the time and

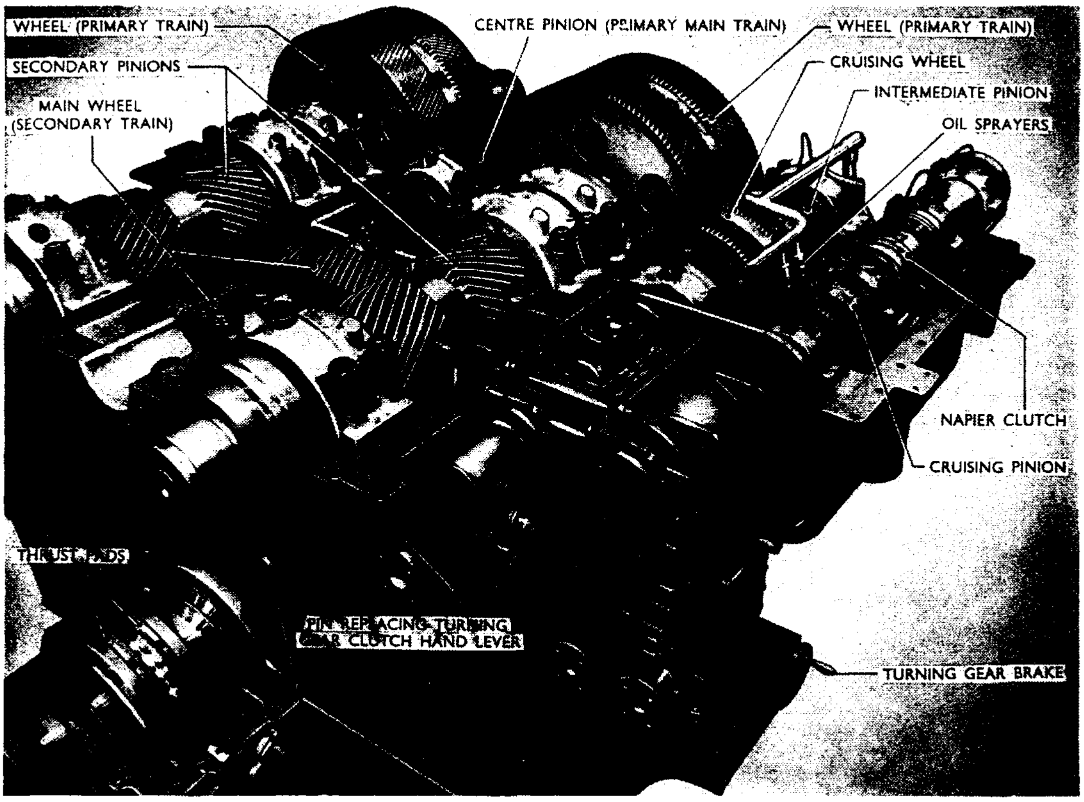


FIG. 1—Y.100 GEARS

cost of production. Both induction hardening and nitriding result in very much less distortion and the load carrying capacity of gears hardened by these processes is being investigated by A.V.G.R.A. Although this work is not yet complete, the results of some recent full-scale gear tests are given and recent advances in the technique of induction hardening are described.

PART I

POST-WAR NAVAL PROPULSION GEARS—DESIGN, TRIALS AND SERVICE EXPERIENCE

' Daring ' Class, Marks I, II and III

The first ships designed and built after the war were the *Daring* Class destroyers and these were also the first major R.N. ships to be fitted with British double-reduction gears. Three gear designs were fitted :

Mark I

A dual tandem articulated design, with hobbed and shaved double helical gears and integral main thrust block forward of the main gear wheel. The gear loadings were 90-100 K (Lloyd's K factor). Some details of this design were given by Page[7].

Mark II

This design was similar to Mark I but the gear loadings were higher (up to 130 K) and a separate main thrust block was fitted close to the after end of the gear case.

Mark III (H.M.S. Diana)

This design had single helical, carburized, hardened and ground pinions and primary wheels and an air hardened main wheel in a dual tandem, articulated arrangement. The gears were designed and manufactured in Switzerland. The gear tooth loadings were up to 260 K.

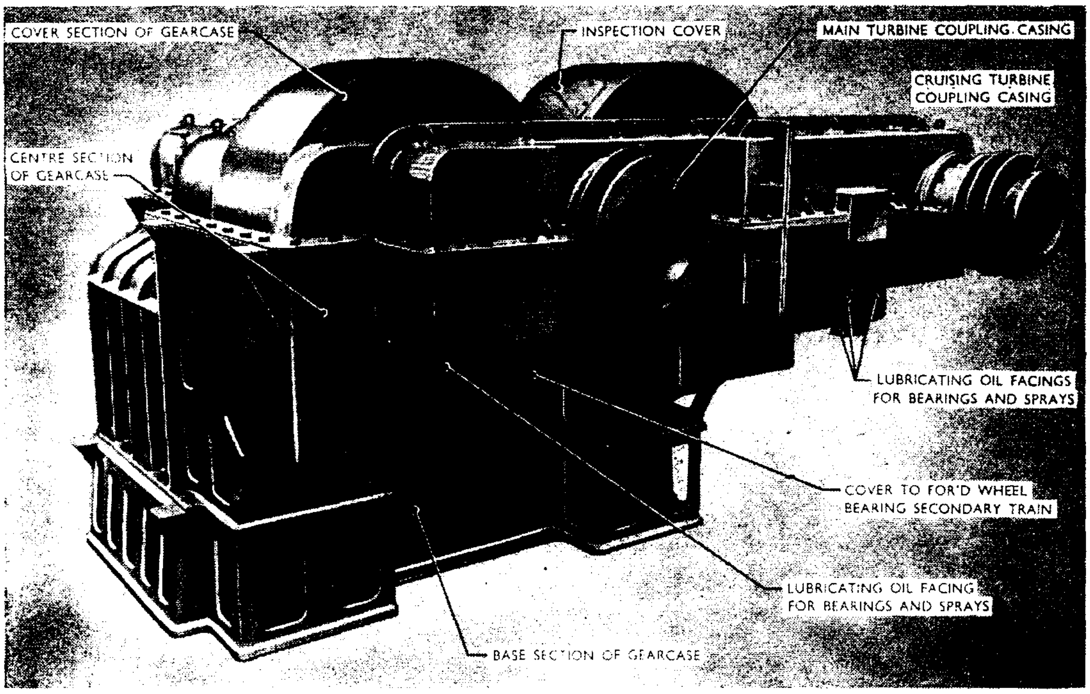


FIG. 2—Y.100 GEARCASE—VIEW FROM FORWARD END

TABLE I.—*Y.100 Gears. General design data*

	Main turbine				*Cruising turbine*			
	Primary		*Secondary*		*Primary*		*Secondary*	
	Pinion	*Wheel*	*Pinion*	*Wheel*	*Pinion*	*Wheel*	*Pinion*	*Wheel*
Number of gear elements	1	2	2	1	1	1	1	1
Number of teeth	37	150	34	211	47	102	37	150
Pitch circle diameter, in. ...	8·508	34·492	12·604	78·22	8·832	19·168	8·508	34·492
Face width at pitch line, in.	12·3 + 2¾ gap Double helical		18·1 + 4 gap Double helical		6·56 + 2¾ gap Double helical		12·3 + 2¾ gap Double helical	
Tangential load/Unit width:								
Full power, lb/in. ...	1,582		2,935					
Full cruising power, lb/in.					869		1,044	
Reduction ratio	4.05		6.21		2·17		4·05	
Total ratio	25·16				54·60			
K Factor:								
Full power	232		270					
Full cruising power ...					144		153	
Helix angle, deg. ...	30		30		32		30	
Normal pitch	0·625		1		0·5		0·625	
Normal pressure angle, deg.	16		16		16		16	
Addendum ratio, pinion/wheel	60/40		60/40		60/40		60/40	
Material Pinion	En.26 70-75 tons u.t.s./sq. in.		En.26 70-75 tons u.t.s./sq. in.		En.26 70-75 tons u.t.s./sq. in.		En.26 ... 70-75 tons u.t.s./sq. in.	
Wheels ...	En.30 60-65 tons u.t.s./sq. in.		En.30 60-65 tons u.t.s./sq. in.		En.30 60-65 tons u.t.s./sq. in.		En.30 60-65 tons u.t.s./sq. in.	
Method of manufacture ...	Hobbed and shaved							

Although pitting of the main wheel teeth was experienced during the shore trials of the first design, all these gears have given satisfactory service. They will not be discussed further here but are referred to again later in the paper when comparative sizes and weights are considered.

The Y.100 Gears

These gears were fitted in the first post-war anti-submarine frigates, the twin-screw *Whitby* Class and the single-screw *Blackwood* Class vessels. In both classes it was necessary to make very substantial reductions in the weights and sizes of all items of machinery and this requirement had a very considerable effect on the gear design.

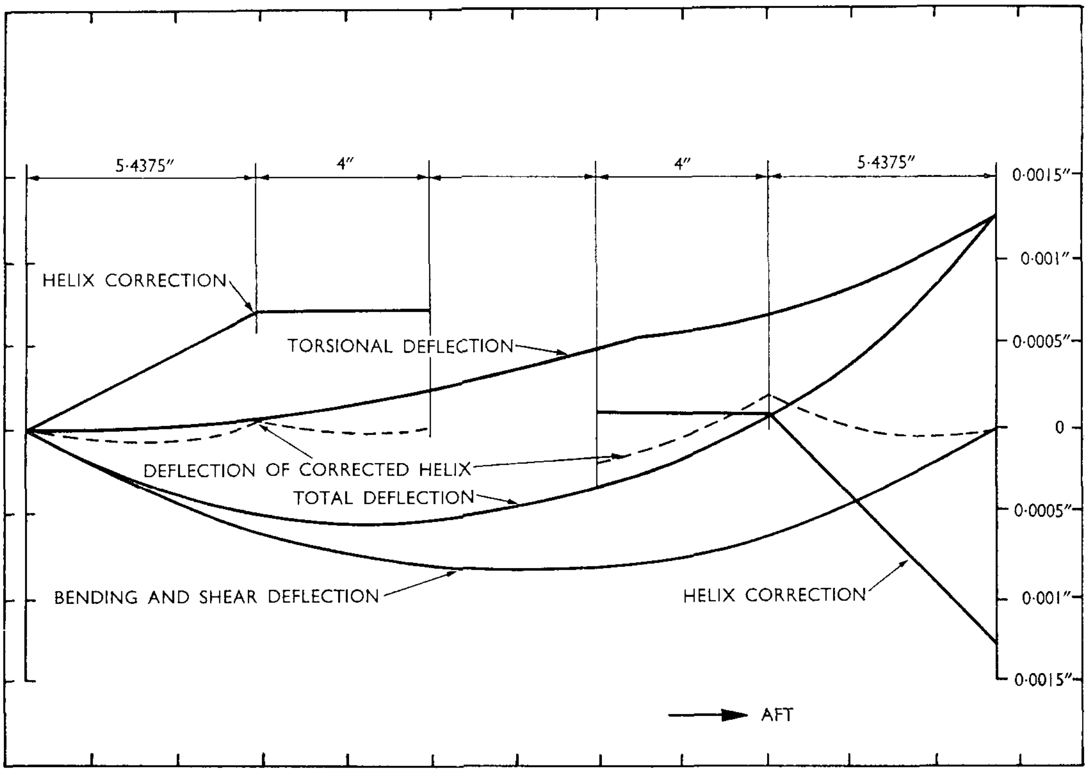


FIG. 3—Y.100 GEARS—ORIGINAL HELIX CORRECTION AS APPLIED TO SECOND REDUCTION PINION

Lack of manufacturing capacity and experience of the production of hardened and ground marine propulsion gears in the U.K. made it necessary to use through hardened, hobbed and shaved materials. At the same time, for a similar duty, the Royal Canadian Navy decided to use carburized, hardened and ground gears and to set up the necessary manufacturing facilities in Canada. Their very successful experience with these gears has been reported by Nicholson[8].

Design Details

The arrangement of the Y.100 Mark I gearing is shown in FIGS. 1 and 2 and general design data are given in TABLE I. The gears transmit 15,000 s.h.p. with a reduction ratio of 5,750 : 225 (cruising gears 8,400 : 154).

The drive from the main turbine was transmitted through double helical, double-reduction, dual tandem, articulated gears and that from the cruising turbine through an automatic clutch and an additional set of gears to the outboard primary wheel. The clutch was situated between the cruising turbine intermediate shaft and primary pinion (see FIG. 1). Its purpose was to disengage, automatically, the cruising turbine at approximately 30 per cent full power when power was increased and to re-engage automatically on reduction of power to this value[9].

To meet the requirements of minimum size and weight the tooth loadings were raised to 270 K for the secondary gears and 230 K for the primaries, i.e. between two and three times higher than those previously used in R.N. propulsion gears of comparable power.

To withstand these loads the materials chosen by the Admiralty were En 26 ($2\frac{1}{2}$ per cent nickel-chromium molybdenum steel) oil quenched and tempered to 70-75 tons for the pinions and En 30b ($4\frac{1}{4}$ per cent nickel-chromium molybdenum steel) air hardened and tempered to 59-67 tons for the wheels, these being the hardest materials that could be satisfactorily hobbed and shaved.

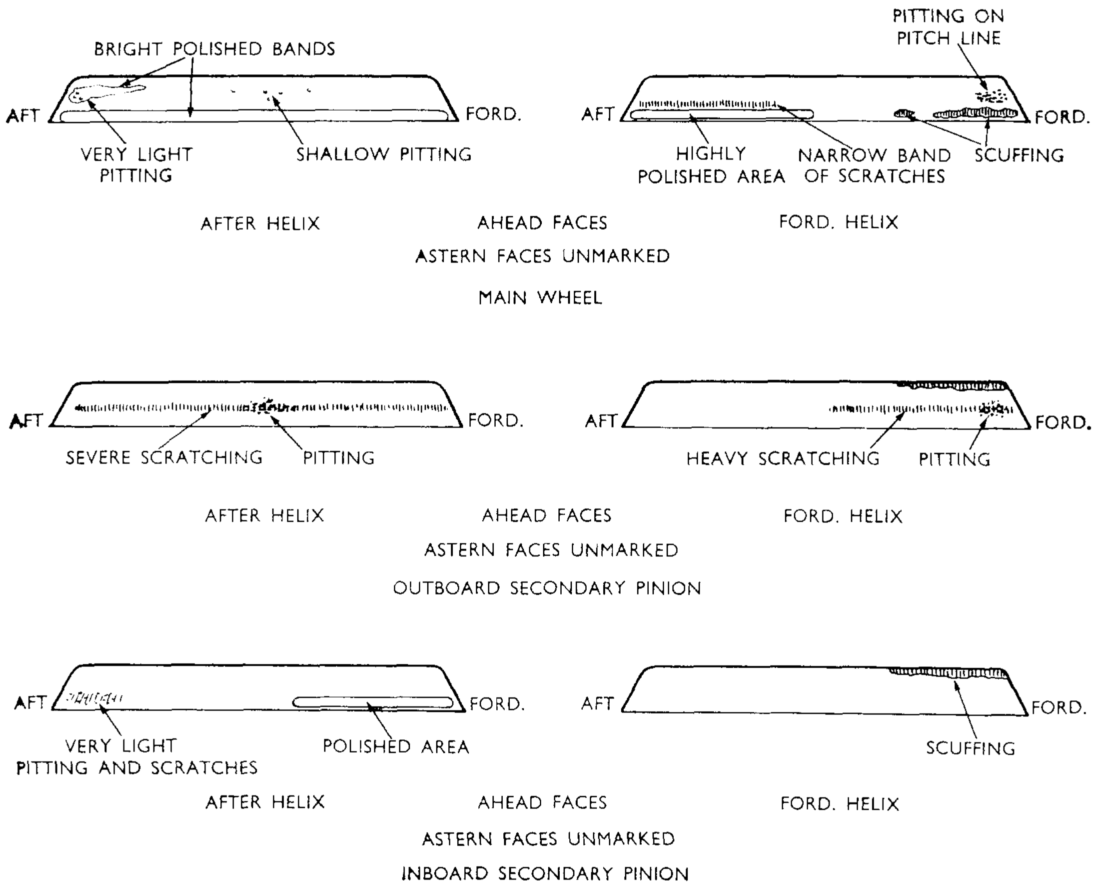


FIG. 4—Y.100 GEARS—CONDITION OF SECONDARY GEARS AFTER 7½ HOURS AT FULL POWER

At the time this choice was made there was no direct evidence in favour of this combination of materials, for although full-scale gear tests had been planned they had not yet been carried out. However, disc tests[5] of other materials indicated that surface load carrying capacity increased with the square of the tensile strength and the choice of materials was made on this basis.

The first three series of S.G.B. (steam gunboat type) gear tests, carried out at Pametrada and subsequently reported by Newman[6] had shown that helix correction designed to compensate for the full load distortion could provide a considerable increase in load carrying capacity. Helix corrections were therefore adopted for the secondary pinions of the Y.100 gears and details of the original corrections are given in FIG. 3. It was appreciated that the rather large normal pitch (1 in.) of the secondary gears might cause a tendency to scuff but E.P. additive lubricating oils were then becoming available and could be used if required.

The gearcase was a fabricated steel structure with steel covers and cast steel bearing housings. Whitemetal bearings in thick steel shells, equipped with thermocouples and mercury-in-steel thermometers, were fitted. The drive from each turbine was transmitted through nitrided gear type, fine tooth couplings. Gear type fine tooth couplings were also fitted at the after end of each quill shaft connecting the primary wheels and secondary pinions. The axial position of the main wheel was determined by the main thrust bearing, the secondary pinions being free to position themselves axially. The position of the outboard primary wheel was located by thrust faces at the ends of the primary wheel forward journal, the primary pinion and inboard primary wheel being free to position themselves axially relative to this located wheel. The weight of the gear set, complete in all respects was 17 tons.

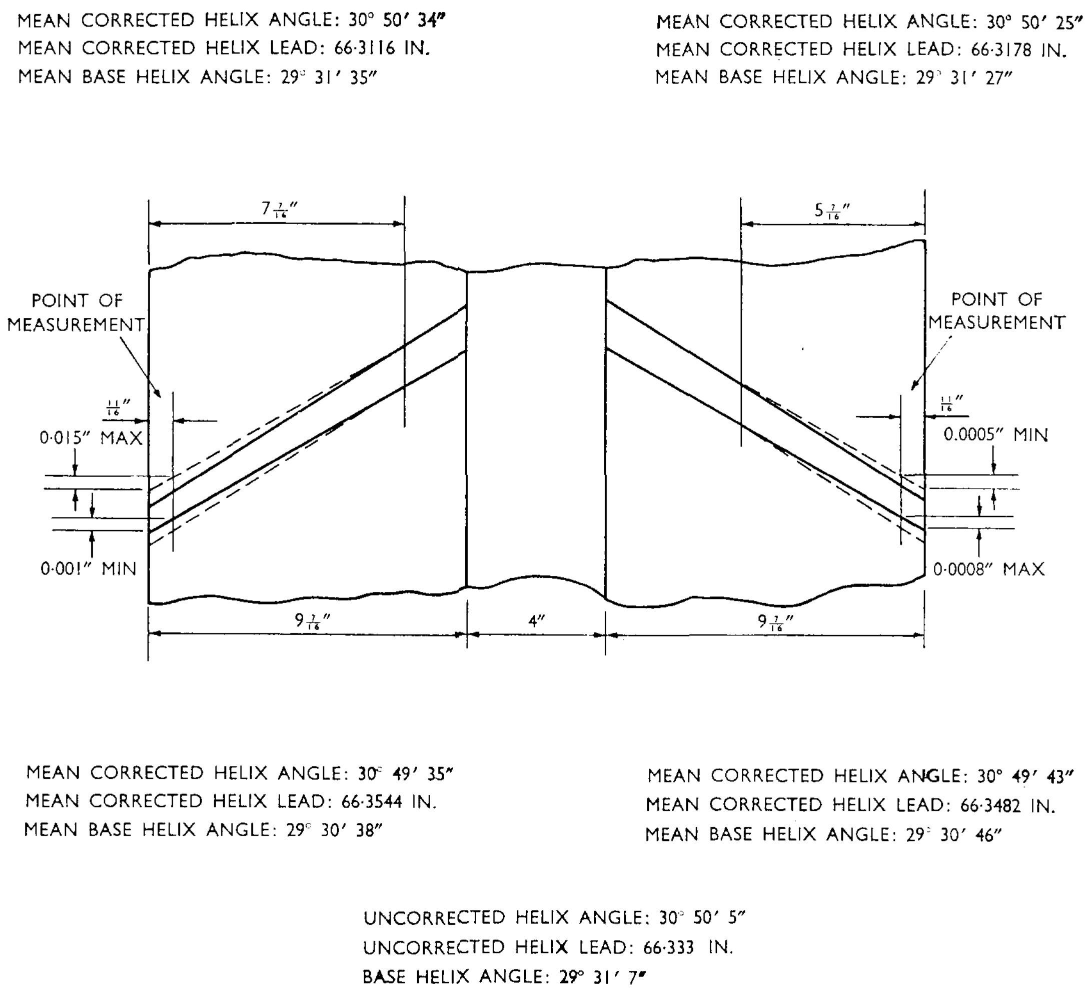


FIG. 5—Y.100 GEARS—MODIFIED HELIX CORRECTIONS AS APPLIED TO INBOARD FINAL REDUCTION

Y.100 Mark I Gears—Shore Trials Experience

The port gear set for the first ship, H.M.S. *Whitby*, was tested at Pametrada in conjunction with the Y.100 machinery and was subsequently installed in the ship. Throughout these trials the lubricating oil used was OM100, which did not contain an E.P. additive.

(a) *Gear Teeth* : Preliminary running of approximately 20 hours at up to 50 per cent torque and 7 hours at 50-100 per cent torque was completed satisfactorily, but after a total of $7\frac{1}{2}$ hours' full power running, deterioration of the secondary gear teeth had begun. Light scuffing over a length of $1\frac{1}{2}$ in. was found at the forward ends of the secondary pinions and main wheel teeth. On the main wheel, the scuffing was towards the roots and on the pinions, it was at the tips of the teeth for about $\frac{3}{16}$ in. down the flanks. Slight pitting had occurred (i) on the main wheel near the pitch line above the scuffed area and at the after end of the aft helix ; (ii) on the outboard secondary pinion at the forward end of the forward helix and near the pitch line of the after helix at a position $4\frac{1}{2}$ in. from the gap ; (iii) on the inboard secondary pinion at the after end of the after helix. This is shown in FIG. 4. The cruising and primary gears were in a satisfactory condition. Further examination and measurement revealed :

(*i*) That shaving of the secondary pinions had produced a profile which was proud at the pitch line and tooth tips with a hollow between, of 0·0003-0·0006 in. in depth. The errors were greatest at the extreme ends of the pinions where most shaving had taken place in applying

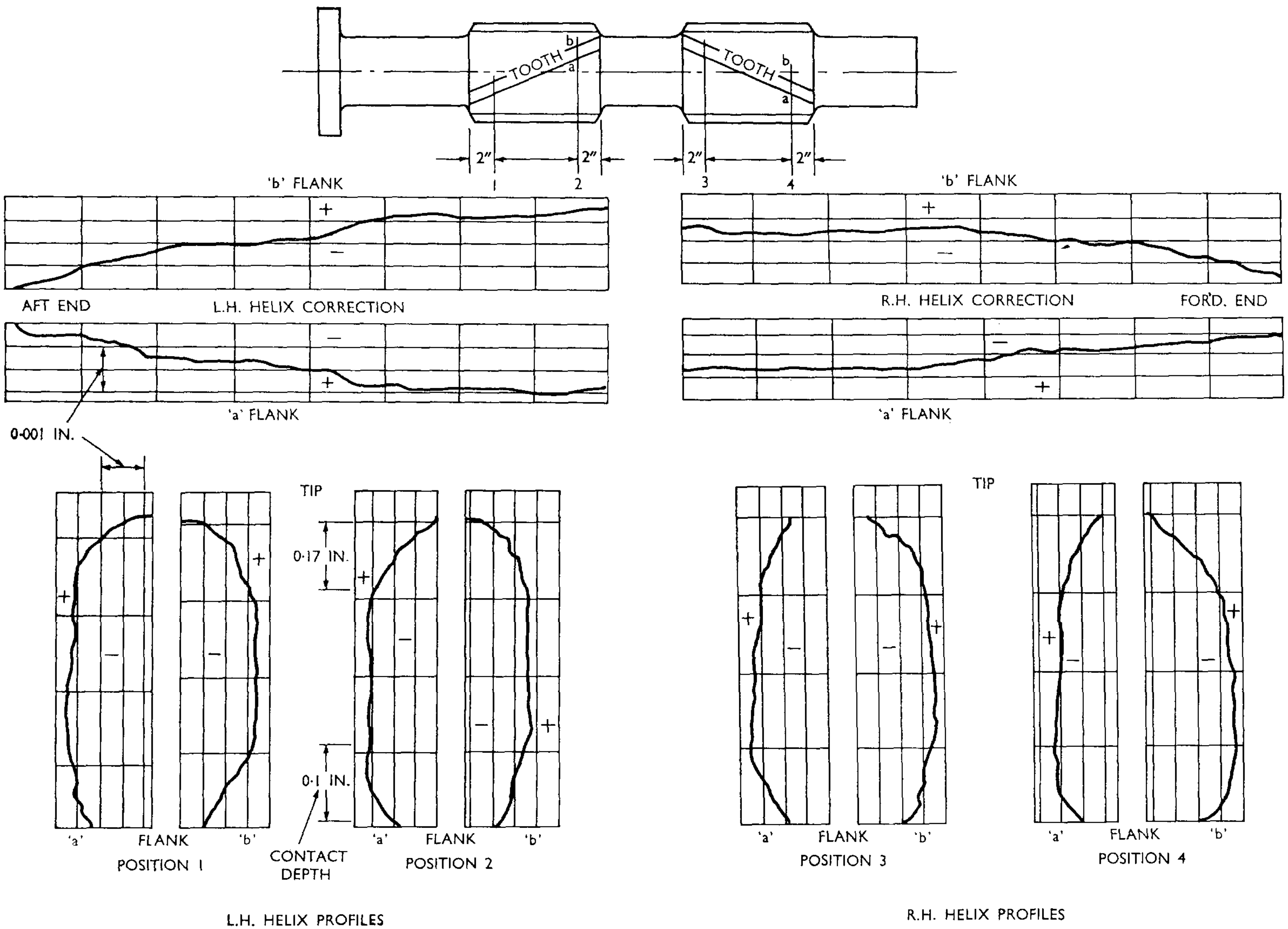


FIG. 6—Y.100 GEARS—RECORDS OF SECONDARY PINION TOOTH PROFILES AND HELIX CORRECTIONS AFTER GRINDING

the helix corrections. It was considered that this had contributed to the pitting and scuffing observed.

(*ii*) That the axial position of the pitting on the after helix of the main wheel coincided with the junction of the corrected and true helices on the secondary pinions. Its position down the tooth corresponded to the pitch line.

Attempts to re-shave the pinions failed, the material having developed a high surface hardness during running. After hand stoning the scuffed areas, the pinions were replaced without further correction, to permit the continuation of main machinery trials. The difference in the pitting and contact marking on the secondary pinions had suggested slight inequality of load sharing and the gears were re-torqued accordingly.

The gearing continued in use until it had completed 166 hours of which about 30 hours were run at between 70 per cent and 100 per cent power. The scuffing practically polished itself out during this running but the pitting continued to spread until the gears were removed, although at no stage could the teeth be described as bad. The primary and cruising gears remained in good condition.,

The complete gear set was returned to the makers for replacement of the secondary train by the ground pinions and shaved wheel originally made for the starboard unit. Grinding of the secondary pinions was resorted to as an expedient, pending the production of suitable shaving cutters.

For the original pinions, modifications to the shaving cutter profiles had been specified on the assumption that they would produce a conjugate shape on the gear being shaved and thus the desired gear tooth profile was only indirectly specified. The original shaving cutters were designed with 0·001-0·0012 in. tip and root relief for both wheels and pinions and it was subsequently found that they left very little, if any, unmodified involute profile. For the new ground pinions, 0·001 in. tip relief, running out tangentially 0·17 in. down from the tip and a root relief of 0·0005 in. running out tangentially 0·10 in. from the end of contact was specified. Profile records of these pinions teeth are shown in FIG. 6.

In addition, both the magnitude and disposition of the helix correction were modified to reduce the tendency to pit at the junction of the true and corrected helices. Details of the revised corrections are shown in FIGS. 5 and 6.

Examination of the original gears had shown evidence of fretting corrosion between the quill shafts and bushes in the bores of the primary wheels and secondary pinions. these bushes being provided only to limit the amount of sagging of the primary wheel and secondary pinion assembly during fitting. It was thought that interference at these positions and hence restriction of the gears within their bearing clearances, might have contributed to the pitting at the forward ends of the forward helices of the pinions and the clearance was therefore increased from 0·008 in. to 0·030 in.

Trials were resumed and the new secondary train gave a satisfactory performance, successfully completing 50 hours at full power and short periods at up to 130 per cent torque. Helix correction was considered justified and the tooth marking tended to corroborate the theoretical deflexion curves. Measurements of the gearcase distortion under load indicated that the forward secondary pinion bearings were slightly more rigidly supported than the after bearings. This would tend to concentrate the loading towards the forward ends of both helices and could have been contributory to the scuffing and heavier pitting observed in the initial trials and also to the slightly heavier loading at the forward ends of the forward helices which appeared to persist during the final trials with the ground pinions. The effects of distortion in this design have been analysed and reported in detail by Waterworth[10].

TABLE II—*Y.100 Gears. Comparative dimensions and loadings of original and reduced length bearings.*

Bearing	*Journal diameter, in.*	*Bearing length*		*L/D ratio*		*Loading modified, lb/sq in.*
		Original, in.	*Modified, in.*	*Original*	*Modified*	
Cruising primary pinion	4	4	1½	$\frac{1}{1}$	$\frac{1}{2 \cdot 66}$	500
Main primary pinion	6	6	3¾	$\frac{1}{1}$	$\frac{1}{1 \cdot 66}$	—
Cruising primary wheel	6	6	3¾	$\frac{1}{1}$	$\frac{1}{1 \cdot 66}$	460
Cruising second pinion	6	6	3¾	$\frac{1}{1}$	$\frac{1}{1 \cdot 66}$	460
Main primary wheel	8½	7½	3	$\frac{1}{1 \cdot 14}$	$\frac{1}{2 \cdot 8}$	430
Main second pinion	10	8	7	$\frac{1}{1 \cdot 25}$	$\frac{1}{1 \cdot 4}$	415
Main wheel	15	11¾	11¾	$\frac{1}{1 \cdot 28}$	$\frac{1}{1 \cdot 28}$	220

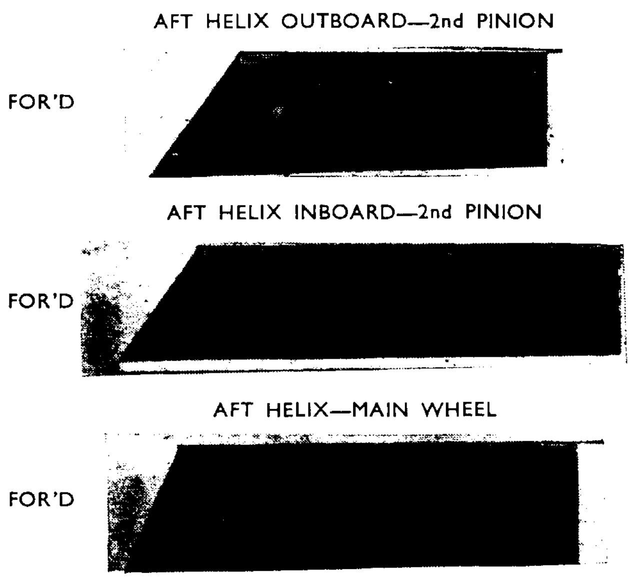


FIG. 7—H.M.S. TORQUAY (Y.100 GEARS)—SELOTAPE RECORDS SHOWING PITTING OF SECONDARY GEARS AFTER ONE HOUR AT FULL POWER

(*b*) *Bearings* : No troubles were experienced with the original journal bearings but after the first series of trials, bearings of reduced length were fitted. Comparative dimensions and loadings are given in TABLE II. The reduction in length was achieved by machining back the whitemetal at each end of the original bearing, leaving a running strip in the centre. These bearings performed excellently at all loads and speeds.

At the conclusion of the trials the gear set was installed in H.M.S. *Whitby* and its subsequent performance is mentioned later in this paper.

Y.100 Mark I Gears—Service Experience

Eighteen starboard and six port gear sets were required for the R.N. ships (six twin-shaft *Whitby* Class and twelve single-shaft *Blackwood* Class frigates) and manufacture was entrusted to a number of firms. Gear-cutting was undertaken by six firms, five of whom produced complete gear sets. In addition, another five firms manufactured gearcases but not gears.

The performance of these gears in the ships has been disappointing. In several sets, quite severe pitting of the secondary pinions occurred during the first few hours at full power. Usually, but not always, this was near the gap, i.e. on the uncorrected portions of the helices and a typical example is shown in FIG. 7. Minor scuffing was also experienced in some sets and it was therefore decided to use E.P. lubricating oils.

The condition of the eighteen starboard and six port sets after contractors' sea trials was as follows :

Starboard Sets (18)	*Port Sets* (6)
10 pitted	1 pitted
4 minor scuffing	3 minor scuffing*
7 satisfactory	3 satisfactory

* Including the port gear set of H.M.S. *Whitby* which had been satisfactory during shore trials.

In all cases the condition of the primary gears was satisfactory. Since sea trials there has been further deterioration in the condition of most of these gears. The port and starboard gears of three of the twin screw *Whitby* Class vessels are still in excellent condition but the remainder (15 starboard and 3 port sets) have all pitted to varying extents. Also, several sets of primary gears now show fine pitting across the full face width in the vicinity of the pitch line. In addition, the secondary gears in a few sets have recently started to pit after being in good condition for a number of years.

In all sets, scuffing has been successfully eliminated by the continued use of E.P. oils.

A number of factors have been suggested to account for these variations in performance :

(i) Gear-cutting errors, particularly in the application of helix correction. The problems encountered in shaving the first set have already been mentioned but it is thought that shaving errors cannot be held entirely responsible, since pitting has also occurred in ground pinions. In some cases there is evidence that excessive helix corrections were applied, resulting in overloading of the uncorrected portions and there is also evidence of excessive tip relief. It is perhaps significant that the successful gears of two of the three *Whitby* Class ships mentioned above were manufactured by the same firm and, in addition, the main gear wheels were lapped and not shaved. In cases where a firm manufactured several gear sets the performance of the later sets has invariably been better than that of the earliest sets made by that firm.

(*ii*) Defects occurred mainly in the starboard sets whereas the port sets were comparatively trouble free. The gears are identical but the directions of rotation are different, the starboard sets running with the apices of the helices leading. With trailing apices, the helix corrections have an effect similar to that of end relief, in reducing the impact on entering mesh and it is possible that this is related to the better performance of the port gear sets. Recently, Boron and Welch[11] described experience with large primary gears, running with trailing apices, which suffered from heavy loading at the gap and they proposed a theory to explain the phenomena. This effect was not experienced in the Y.100 secondary gears and it is possible that other factors were more important.

(*iii*) Gearcase distortion, measured during the shore trials at Pametrada, tended to increase the loading at the forward ends of both helices of the secondary pinions. As in the shore trials, all round chocking of the gearcase was employed in the ships but the greater flexibility of the seatings in the ships may have resulted in greater distortion and resultant maldistribution of load.

(*iv*) The Y.100 gear materials were eventually tested in the S.G.B. gear test rig at Pametrada and, as described by Newman[6] the results were disappointing. It appeared that this combination of materials was prone to scuffing and showed a pitting resistance very much lower than was expected from considerations of hardness and ultimate tensile strength. In fact, later tests showed that the combination En 26/En 9 was more satisfactory in both respects.

(*v*) Andersen and Zrodowski[12] and others have shown how the methods of aligning propeller shafting and main gear wheel can affect internal gear alignment on load and result in local overloading. During the trials of the first two *Whitby* Class ships there was evidence that the starboard forward main wheel journal was lifting into the top half of its bearing during high speed turns to starboard. The cause of this has never been established. Neither has the thermal rise of these gear sets nor the alignment actually achieved in a particular case been checked.

Regrettably, it has not been possible to investigate fully and explain the different behaviour of the gears in different ships. However, it seems reasonable to conclude that for the materials chosen and the standards of accuracy achieved during manufacture, the designed tooth loadings (270 K secondaries, 230 K primaries) were too high. Experience with these ships and the results of full-scale shore trials suggest certain conclusions regarding through hardened, hobbed and shaved gears :

(*a*) With the best available material combination (En 26 pinions and En 9 wheels are now favoured) the maximum permissible loads should not exceed 200 K for primary gears and 160 K for secondaries. Even at these loadings the margins of safety from pitting will not be great and first class manufacture (Grade A1 B.S. 1807 : 1952) and installation is necessary.

(*b*) For warship propulsion gears where further reduction in size and weight are required, it becomes necessary to use surface hardened gears. Factors other than tooth strength may then influence the size and geometry of the gears, e.g. spread of turbines, and at present it seems likely that there will be no advantage in using loadings in excess of about 500 K. At such loadings surface hardened gears can have very substantial margins of safety from failure by pitting or tooth breakage, as indicated by the full-scale gear tests carried out by A.V.G.R.A. and reported by Newman[6] and the excellent performance of the carburized and ground gears fitted in the *St. Laurent* Class frigates of the Royal Canadian Navy and reported by Nicholson[8].

Y.100 Mark II Gears

For the Y.100 Mark II gears fitted in the *Rothesay* and *Leander* Classes and the ships of a number of Commonwealth navies some major changes were made, although the gear tooth geometry and loading were not altered :

(*i*) Carburized, hardened and ground secondary pinions in En 36 steel were fitted

(*ii*) Port and starboard gears were handed so that both ran with the apices trailing

(*iii*) Prefinished medium wall bearings, with intermediate sleeves were fitted and where necessary, these sleeves could be ground eccentrically to obtain correct alignment.

When tested in the A.V.G.R.A. second reduction test rig[6] this En 36 and En 30b material combination showed only a small improvement in load carrying

FIG. 8—Y.E.A.D.1—EXTERNAL VIEW OF GEARBOX

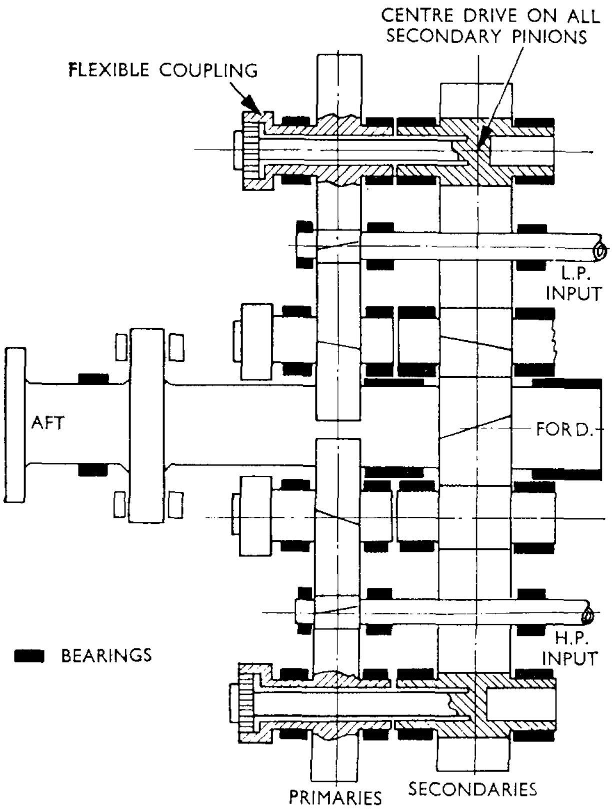


FIG. 9—Y.E.A.D.1—ARRANGEMENT OF GEARS

capacity for minor pitting of the wheel teeth began at 340 K.

However, more than thirty of these gear sets are in service and so far the only damage experienced has been caused by dirt and swarf left in bearing housings and oilways during manufacture. In addition, one of these ships completed two years' service, including approximately 50 hours at 90-100 per cent full power and throughout this period the inboard secondary pinion of the starboard gears had been running in a grossly misaligned condition. So far there is no evidence of damage to the pinion or main wheel. The misalignment has, of course, been corrected.

Nitrided Primary Wheels

One of these gear sets has been fitted for trial purposes, with a carburized and ground En 36 primary pinion and nitrided and ground En 40c primary wheels. The face width of these gears has been reduced to raise the tooth loading to 450 K. This ship will go into service shortly.

Y.200 Mark III Gears

This design is a further development of the Y.100 Mark II and is being fitted in the latest *Leander* Class frigates. In view of the recent occurrence of pitting in the primary gears of some of the original Mark I gear sets, a carburized hardened and ground En 36 primary pinion has been specified. Other modifications have been adopted to permit more thorough flushing of internal oilways to remove dirt and swarf left in during manufacture and, because of difficulty experienced by some manufacturers in achieving satisfactory alignment, adjustable bearings will be fitted. In addition, the cruising gears will be omitted since these ships are not fitted with cruising turbines and the gearcase has been redesigned accordingly.

The Y.E.A.D. I Gears

The 30,000 h.p. Y.E.A.D. I gears (Fig 8) represented the first British attempt to design and manufacture a carburized and ground marine propulsion gear set of large power. These gears and their associated turbines and boiler were extensively tested at Pametrada but were not fitted in any ships. The design was of the double-reduction, dual tandem, articulated type, with single helical,

TABLE III—*Y.E.A.D. 1. gearing. General design data for gears and bearings*

Particulars	*H.P. primary pinion*	*L.P. primary pinion*	*H.P. and L.P. primary wheels*	*H.P. and L.P. secondary pinions*	*Main wheels*
Number of teeth	38	49	210	46	324
Pitch diameter, in.	6·426	8·287	35·515	11·805	83·159
Diameter of addendum circle	6·776	8·636	35·801	12·366	83·610
Diameter of root circle ...	5·977	7·837	35·002	11·086	82·330
Normal pressure angle ...	25 deg.	25 deg.	25 deg.	22 deg. 30 min.	22 deg. 30 min
Centre distances	20·971	21·900	—	47·479	47·479
Helix angle	19 deg. 46 min.	19 deg. 46 min.	19 deg. 46 min.	7 deg. 9 min.	7 deg. 9 min.
Normal pitch, in.	0·5	0·5	0·5	0·8	0·8
Circular pitch, in.	0·5313	0·5313	0 5313	0·8063	0·8063
Face width, in.	8	8	8	13⅛	13⅛
Load/inch of gear face, lb ...	2,370	2,370	2,370	4,340	4,340
K factors	436	353	—	422	—
Bearing size, diameter and length, in.	5·25 × 3·1	5·25 × 3·1	9 × 4·5	10·5 × 6·375	14·5 × 12
Load/sq in. projected area, lb ...	448·5	454·5	427	495	493
Load per bearing, lb ...	7,300	7,400	17,300	32,600	85,800

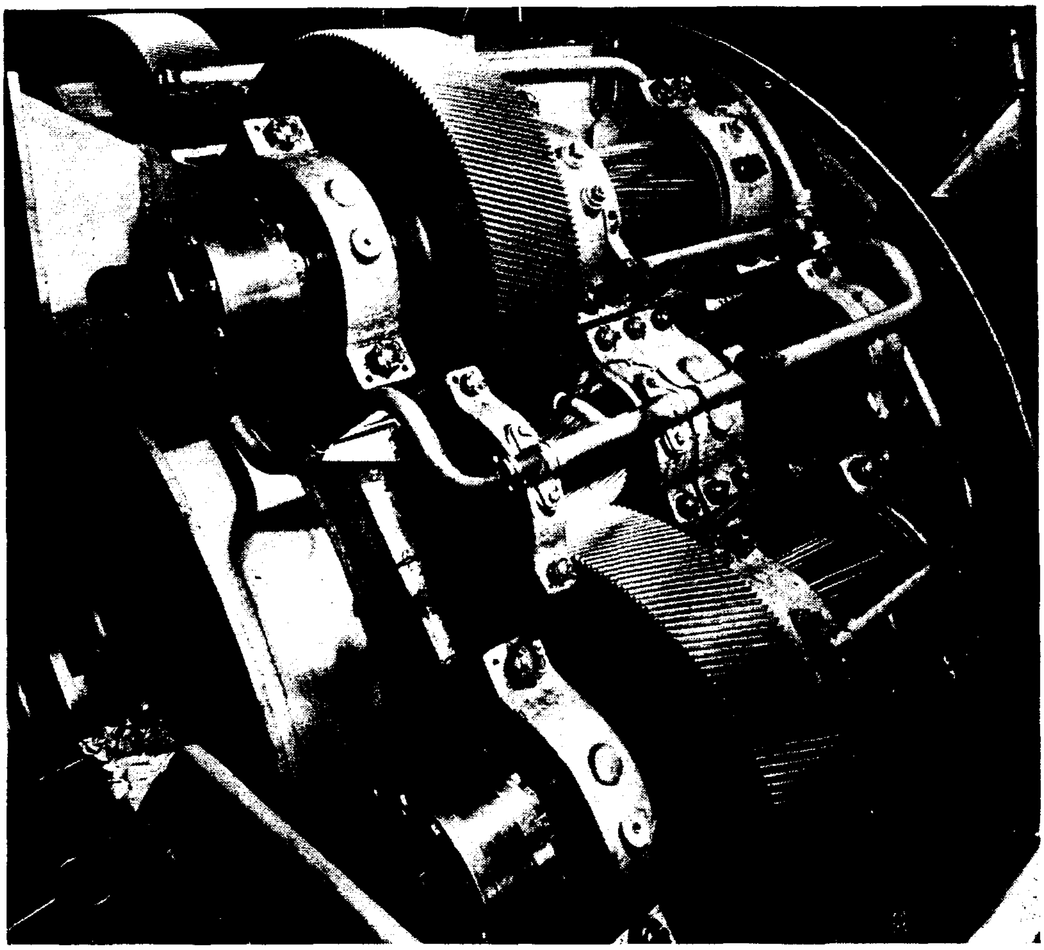


FIG. 10—Y.E.A.D.1—L.P. GEAR TRAIN

En 36a carburized, hardened and ground gears and ran at the following speeds at full power :

H.P. pinion	7,767 r.p.m.
L.P. pinion	6,023 r.p.m.
Main shaft	199·5 r.p.m.

The gears were required to run satisfactorily on OM100 lubricating oil which does not contain E.P. additives, and in view of the scuffing troubles experienced with the Royal Canadian Navy's carburized and ground Y.100 gears[8], it was decided to use somewhat larger pressure angles and smaller tooth pitches. The details are shown in TABLE III.

TABLE IV—*Y.E.A.D. 1. Gearing lubricating oil flows, etc., at 100 per cent power.*

Bearings	*Flow gal/min*	*Pressure lb/sq in.*
H.P. intershaft	52	11
L.P. intershaft	43	9
Primary pinion and thrust	69	7
Main wheel	4·6	9
Main thrust	22·3	4·5
Secondary gear sprayers	15·7	9·5
Primary gear sprayers	16·0	9·0
Total oil to gearcase	222·6	—
Distribution manifold pressure ..	—	13
Lubricating oil pump discharge ..	—	22·5
Lubricating oil cooler pressure drop	—	2
Cooler inlet temperature	153 deg. F.	
Cooler outlet temperature	120 deg. F.	

The layout of the gearing is shown in FIGS. 9 and 10. It will be seen that the primary gears are situated at the after end of the gearcase and the Michell thrust block, which was a separate unit, was bolted to the after end. The reason for this arrangement was associated with the use of ' three area support '. The major forces to be resisted are those from the secondary gear tooth reactions and it is desirable that these should be transmitted to two areas of support abreast the main gear wheel via transverse strength members. It is convenient for the third area to incorporate the seating for the integral main thrust block, situated in the arch formed below the first reduction gears. If the first reduction gears are situated at the forward end the thrust block becomes inaccessible. A disadvantage of the Y.E.A.D. 1 type of arrangement is the need for a long high-speed shaft to the primary pinions and the use of an additional bearing.

Transmission to the four secondary pinions was made via fine tooth couplings and quill shafts (FIG. 11) and an important feature of the design was the centre drive to the secondary pinions, intended to counterbalance the torsional deflexion by the bending deflexion and achieve even loading across the face width at full power. The fabricated steel gearcase was constructed in two parts, the lower part being a heavily ribbed box structure which carried the main wheel bearings (FIGS 12 and 16). The upper half was built in the form of three bridges over the main wheel shaft and the forward end of the thrust block, with the H.P. bearings on the outboard side of the bridges and the L.P. bearings on the inboard side. Each set of steel bearing housings was cast in one piece and recessed and welded into the bridges (FIG. 13). Aluminium covers were fitted. The weight of the gearbox, complete, was 27½ tons.

Gear Proving Trials

Gear proving trials were carried out over a range of powers from ' light load ' to full power, with instrumentation arranged for measuring bearing temperatures, conditions in the lubricating oil system and gearcase distortion. The lubricating oil used throughout the trials was OM100 and did not contain E.P. additives.

The gears performed very satisfactorily throughout the trials and the gear tooth surfaces remained in excellent condition with indications of full face contact at full power. Originally some of the high speed bearings were unreliable and ran hot but no further trouble was experienced after their diametral clearances had been increased from 0.007 in. to 0.012 in. Typical records of lubricating oil pressures, temperatures and flows are given in TABLE IV.

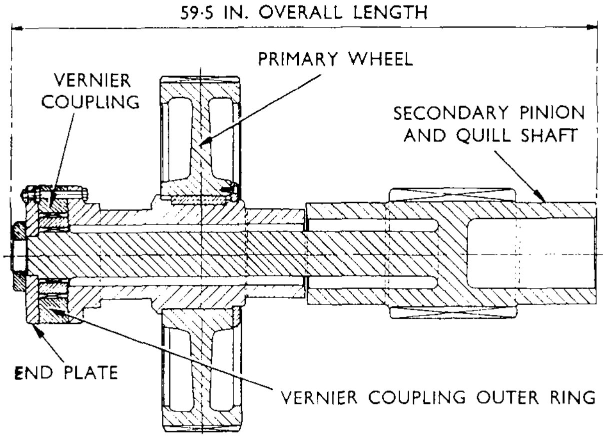


FIG. 11—Y.E.A.D.1 GEARING—ASSEMBLY OF PRIMARY WHEEL AND SECONDARY PINION

To measure distortion under load a rigid frame was built up around the gearcase from the test house floor. It was entirely independent of the gearcase and machinery seatings and carried dial gauges by means of which the absolute movement of several points on the gearcase could be determined.

FIG. 14 shows a portion of the distortion frame, the dial gauges and the measuring stalks which were screwed into tapped holes on the bearing keeps

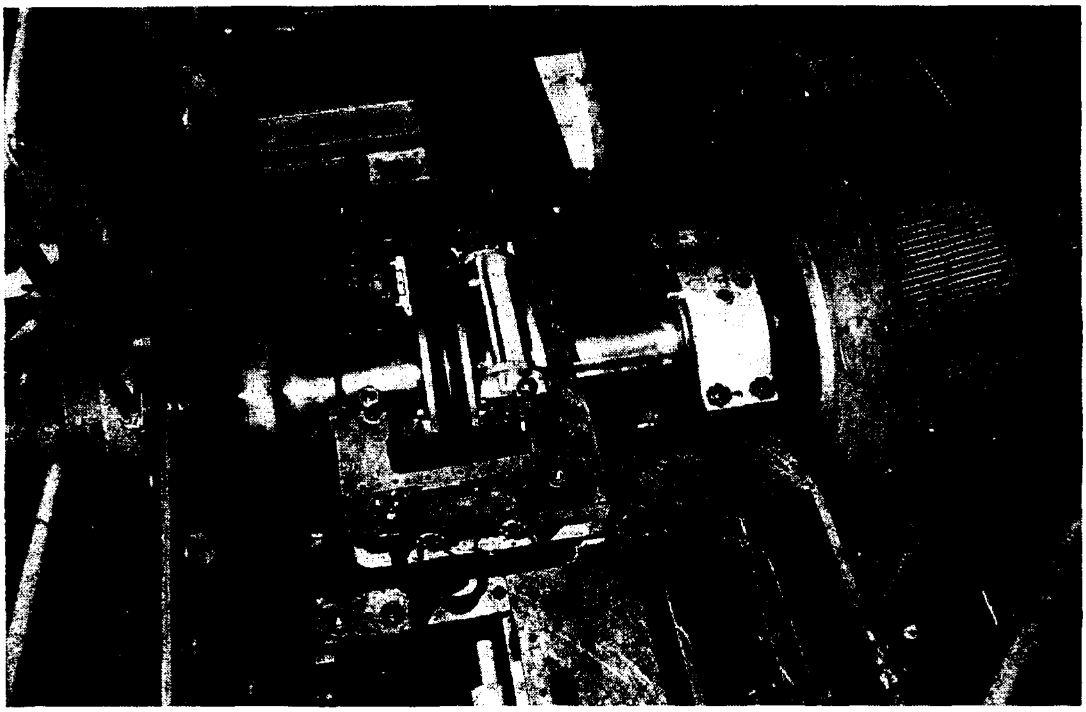


FIG. 12—Y.E.A.D.1—GEARCASE WITH TOP HALF REMOVED

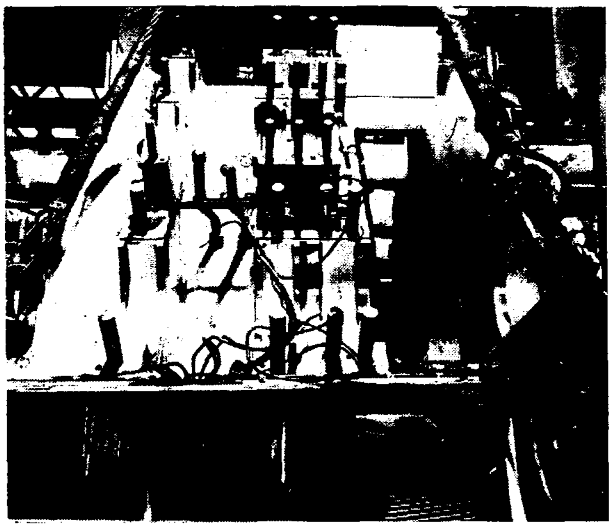


FIG. 13—Y.E.A.D.1—BEARING HOUSING AND BRIDGES

and passed out through clear holes in the gearcase covers sealed by brass bellows (0.005 in. thick). The end of each stalk had two flat faces at right angles to each other against which the dial gauge spindles bore, these faces being machined in each case so that they were respectively horizontal and vertical.

Just before the end of each run, all the dial gauges were set to zero and the engines were stopped in about one minute. Immediately the turbines had come to rest the dial gauges

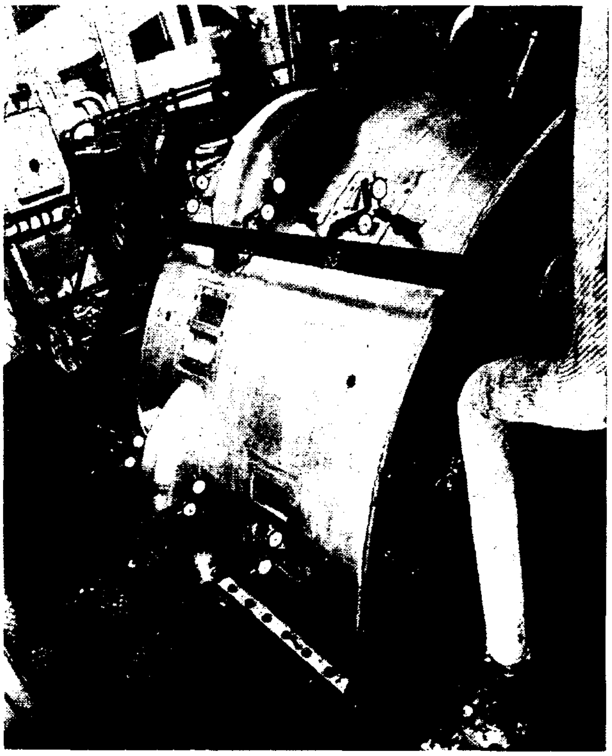


FIG. 14—Y.E.A.D.1—L.P. SIDE SHOWING DISTORTION FRAME AND DIAL GAUGES

were read again. The engines were than taken up to power once more in about one minute, dial gauge readings were taken and the procedure was repeated to obtain check readings. In this way the distortion of the gearcase caused by bearing loads alone was obtained while the thermal distortion effect was practically eliminated. In all, the vertical and horizontal movements of twenty points around the gearcase were measured and are shown, for the full power condition, in column 3 of FIG. 15. Column 1 of this figure shows the bearing loadings at each frame, the resultant of the loads in the top half of the gearcase being shown with a radius from the main gear wheel axis to define its position. An analysis of the effects of these forces on the complex structures of the gearcase (Fig. 16) would be lengthy

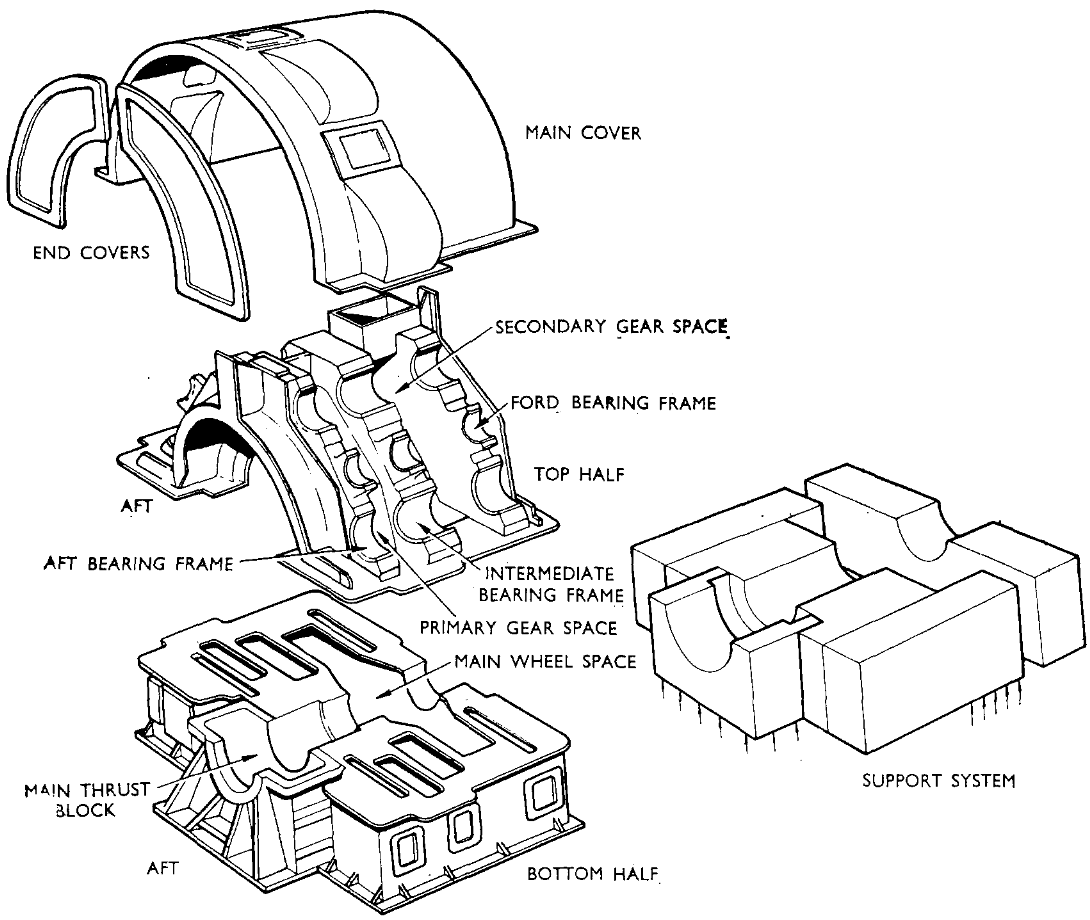


FIG. 16—Y.E.A.D.1.—MAIN STRUCTURAL COMPONENTS OF GEARCASE

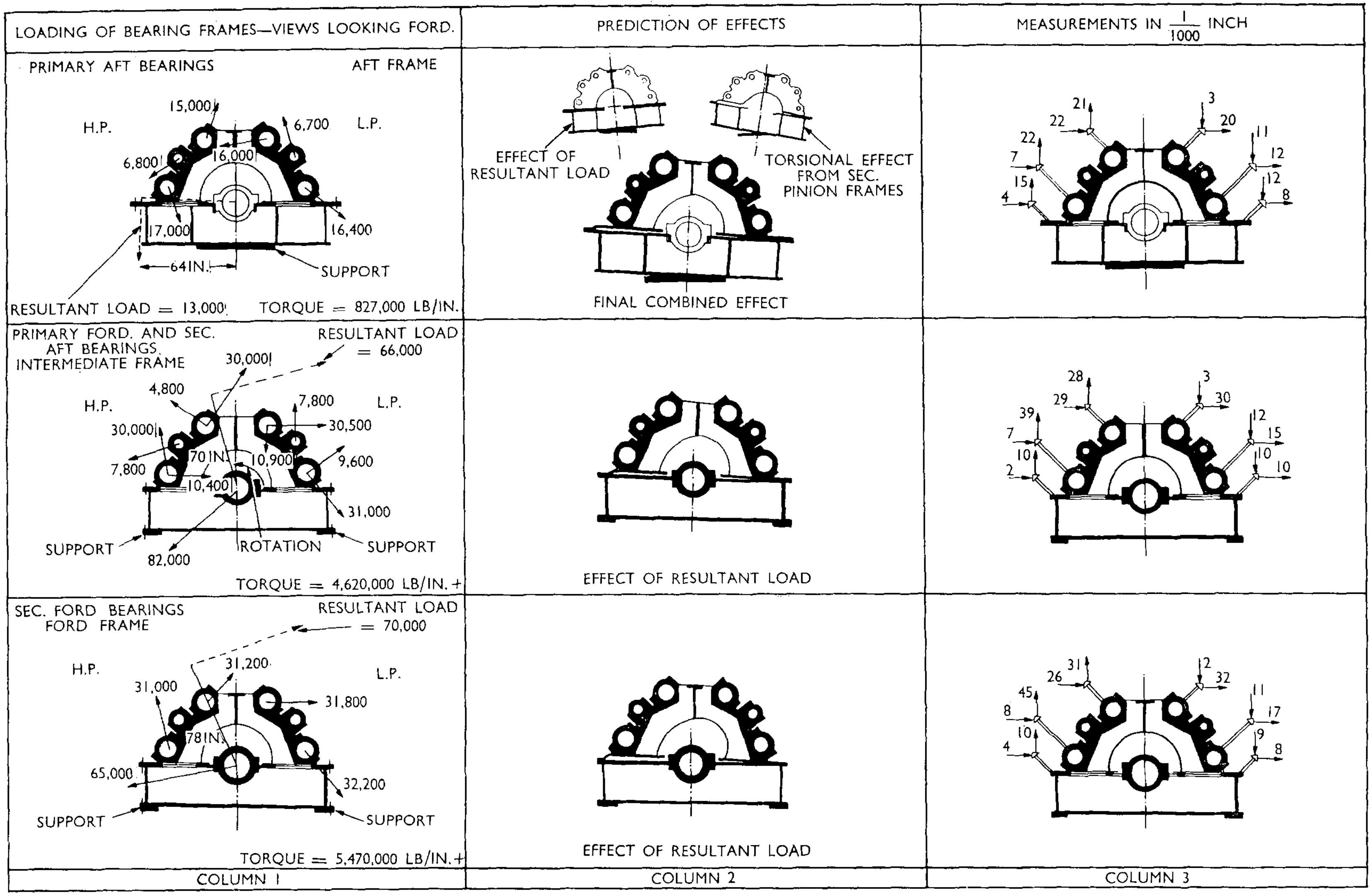


FIG. 15—Y.E.A.D.1—ANALYSIS OF DEFLECTION TESTS

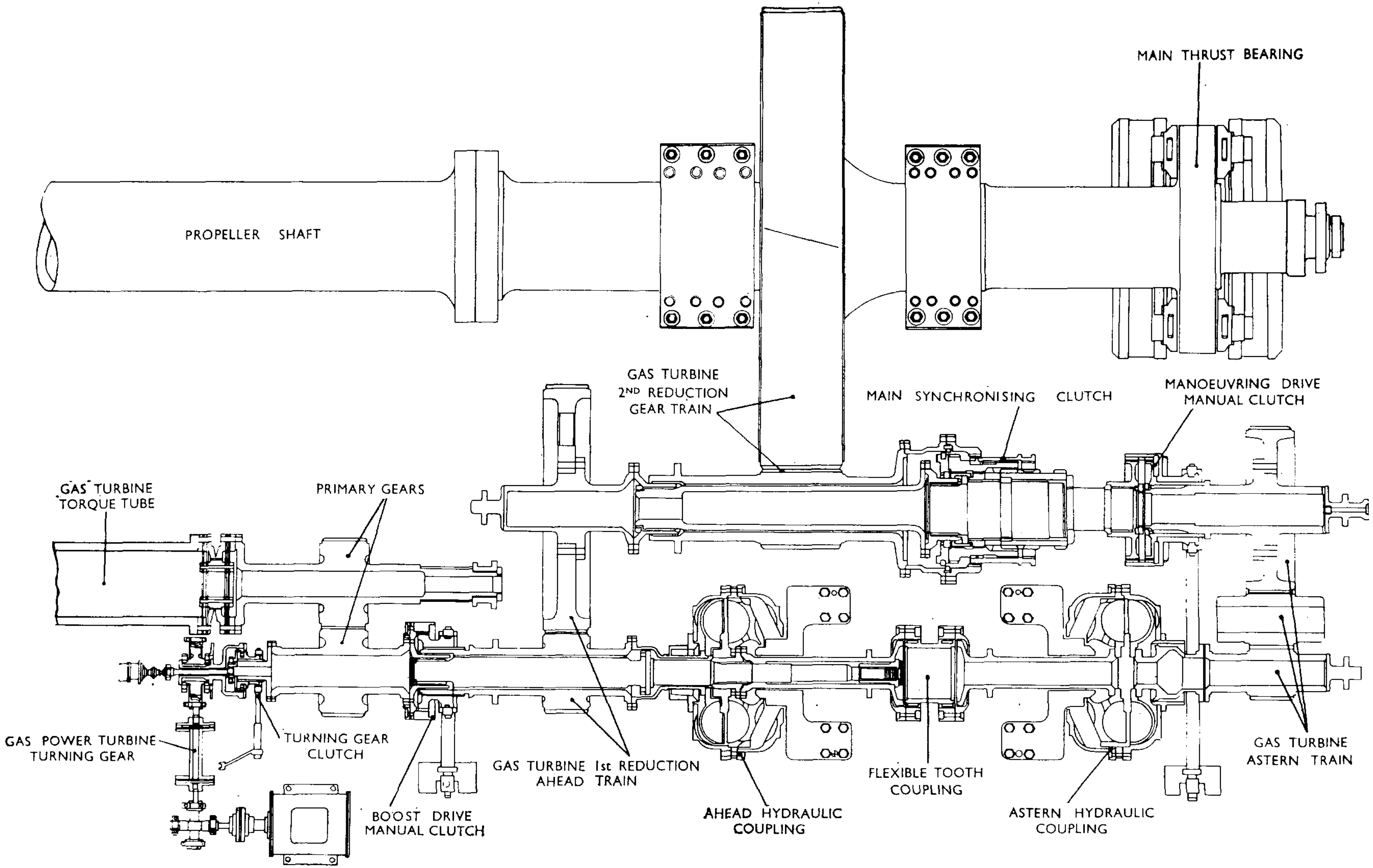


FIG. 17—Y.102A—PORT GEARCASE—INBOARD GAS TURBINE GEAR TRAIN

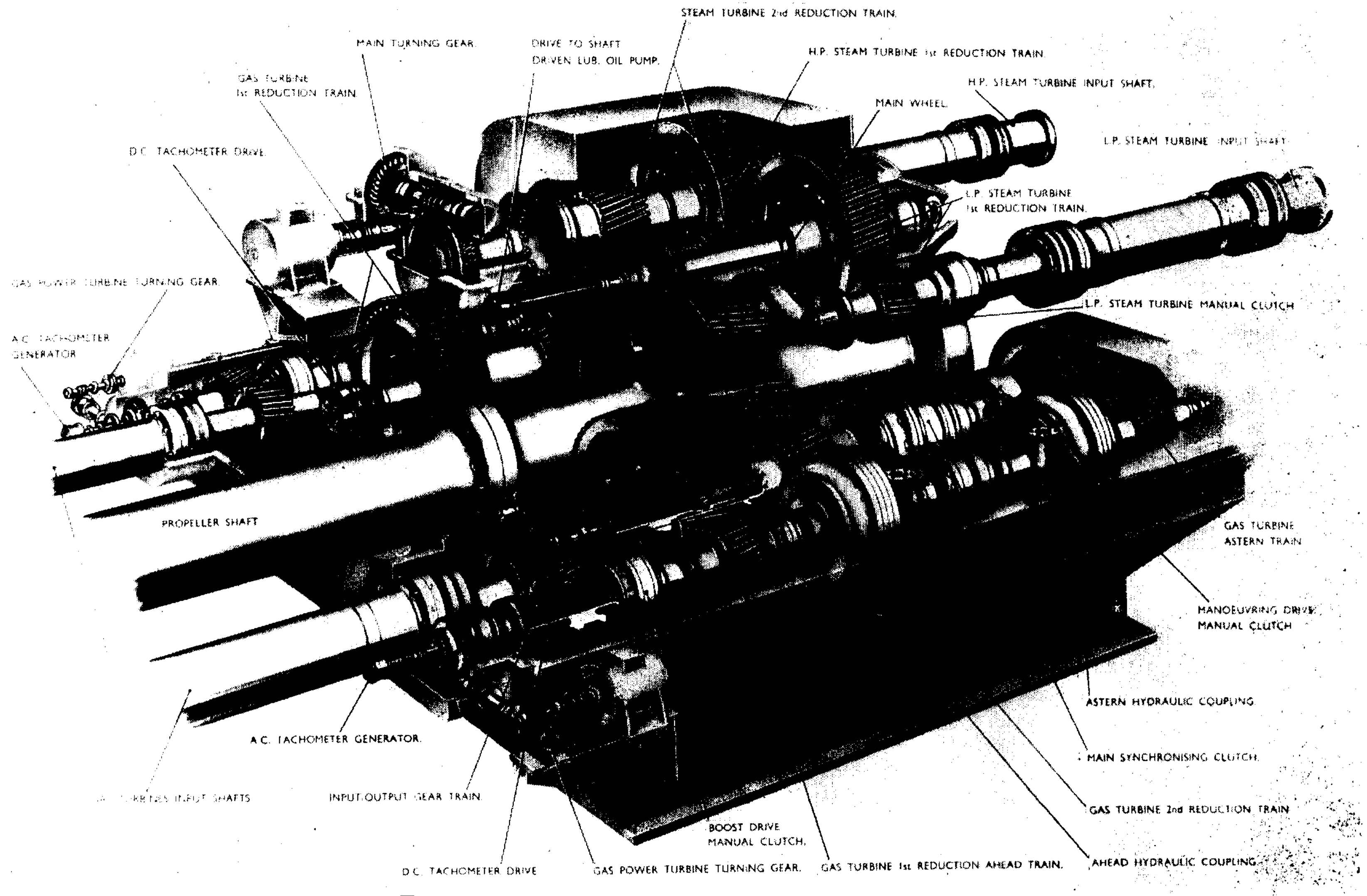


FIG. 19—Y.102A—SECTION VIEW OF GEARBOX, PORT SET

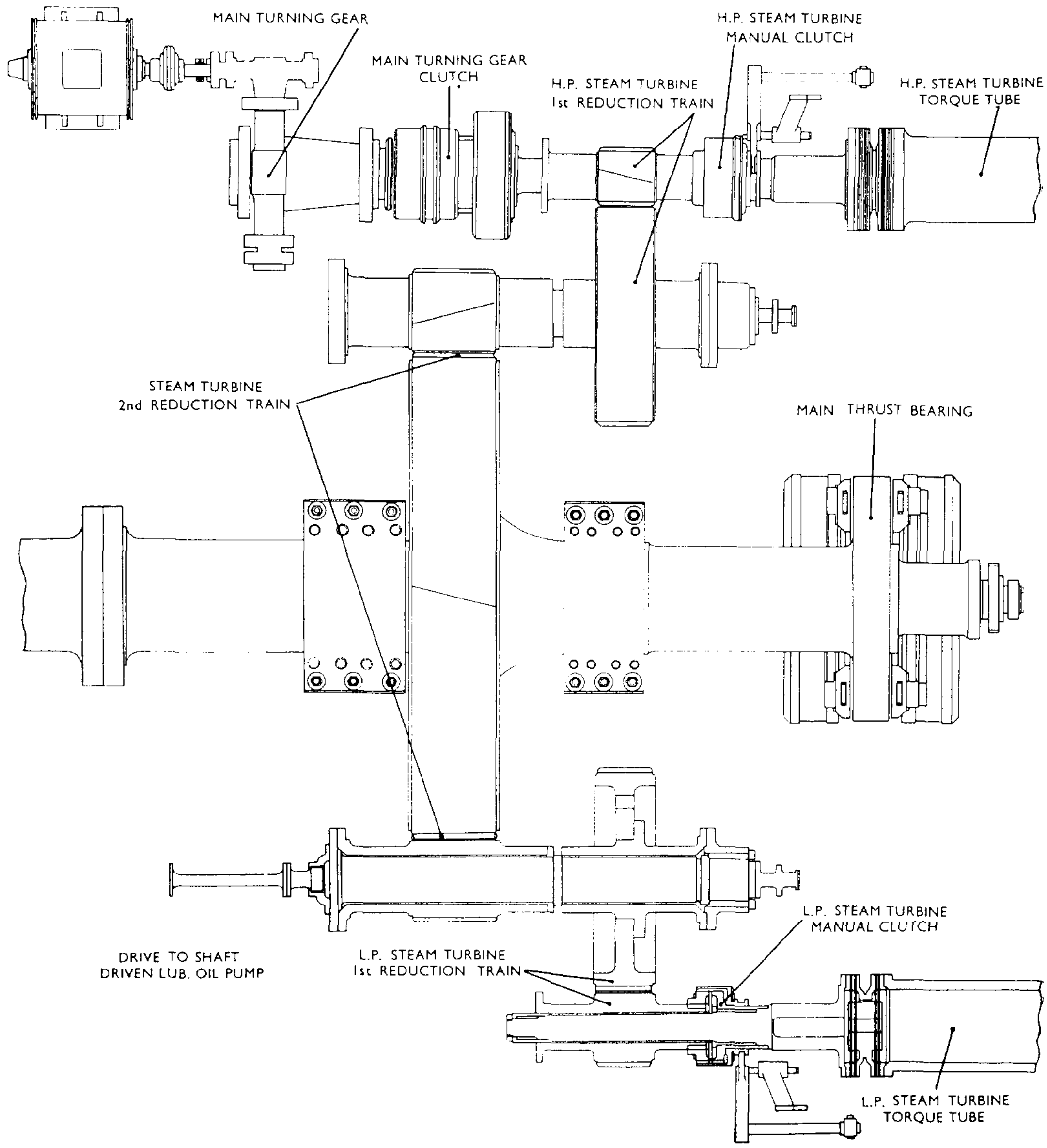


FIG. 18—Y.102A—PORT GEARCASE STEAM TURBINE TRAINS

and uncertain and will not be attempted here. However, visual inspection of the various loads and their directions suggests the effects shown in column 2 of FIG. 15 and these are confirmed by the recorded measurements in column 3.

The Y.102A and Y.111A Gears

The Y.102A and Y.111A gears are fitted in the new *County* Class Guided Missile ships and the *Tribal* Class General Purpose frigates respectively. The main propulsion machinery of these ships consists of steam and gas turbines and the gear designs permit :

(*i*) Ahead and astern operation with the ahead and astern steam turbines

(*ii*) Use of gas turbines to boost the ahead steam turbine power output

(*iii*) Use of gas turbines alone for ahead power

(*iv*) Use of gas turbines alone for ahead and astern operation, i.e. for manœuvring the ship.

The machinery installations of these ships have already been described by

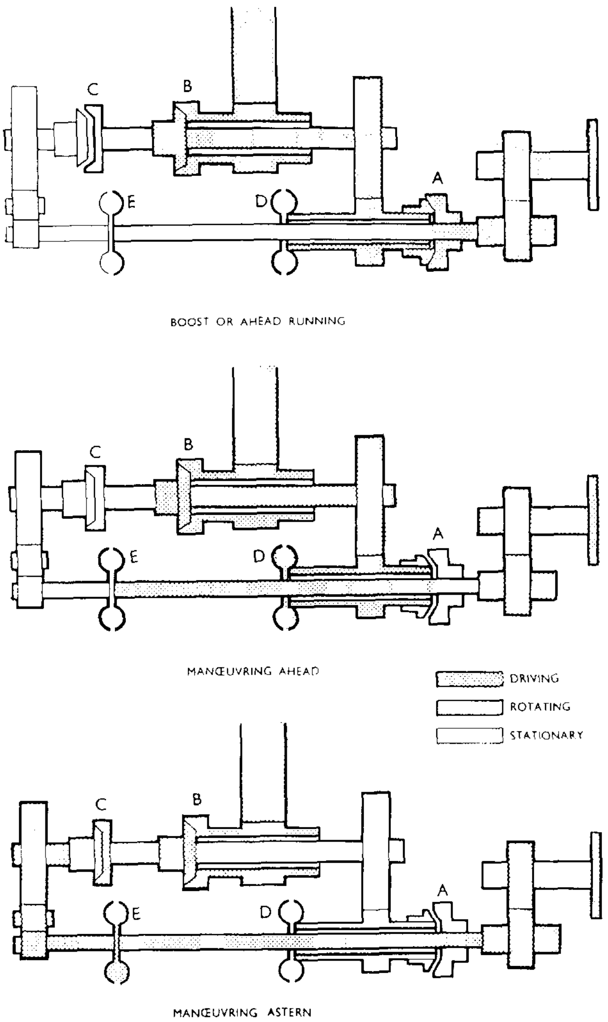


FIG. 20—DIAGRAMMATIC ARRANGEMENT FOR PORT GAS TURBINE GEAR TRAIN

Good and Dunlop[13] who also gave an account of the extensive shore trials carried out on the prototype gas turbines and Y.102 gearbox and of sea trials experience in the first ships of each Class. In order to present a complete picture, some of the information given in their paper will be repeated below but, as far as possible, the authors will confine their remarks to the more detailed aspects of these gears which have not yet been published.

General Description

The general layout of the Y.102A gears is shown diagrammatically in FIGS. 17 and 18 and in a section view in FIG. 19. The steam turbine gear trains and gas turbine boost gear trains are of the double-reduction, tandem articulated type and the manœuvring drive from each gas turbine comprises ahead and astern gear trains with associated hydraulic couplings, manually operated clutches and main synchronizing clutches. The main synchronizing clutches permit the gas turbines to be connected to the propeller shaft when it is rotating in the ahead direction. They engage automatically at synchronism and are then locked into engagement so that they are capable of transmitting ahead or astern torques. A diagrammatic representation of one of the gas turbine gear trains is shown in FIG. 20, from which the various operating functions of the gas turbines can be visualized. Thus, if a gas turbine is required for boost or ahead running only, the drive is taken through the boost drive manual clutch (A) which is engaged, through the first reduction ahead gears to the main synchronizing clutch (B) and thence through the second reduction pinion to the main gear wheel. The manœuvring drive manual clutch (C) remains disengaged.

When required for manœuvring the gas turbine drive is connected to the hydraulic couplings (D) and (E) with the manœuvring drive manual clutch (C) engaged and the boost drive manual clutch (A) disengaged. Oil is supplied to either the ahead or astern hydraulic coupling as required.

FIG. 21 illustrates the Y.111A gearbox, which is similar in principle but simpler, since only one gas turbine and one steam turbine are involved. The steam turbine gear trains are, however, of the dual tandem, articulated type.

Hydraulic Couplings

A diagrammatic arrangement of a pair of hydraulic couplings with their associated control gear is shown in FIG. 22. A sectional arrangement of an

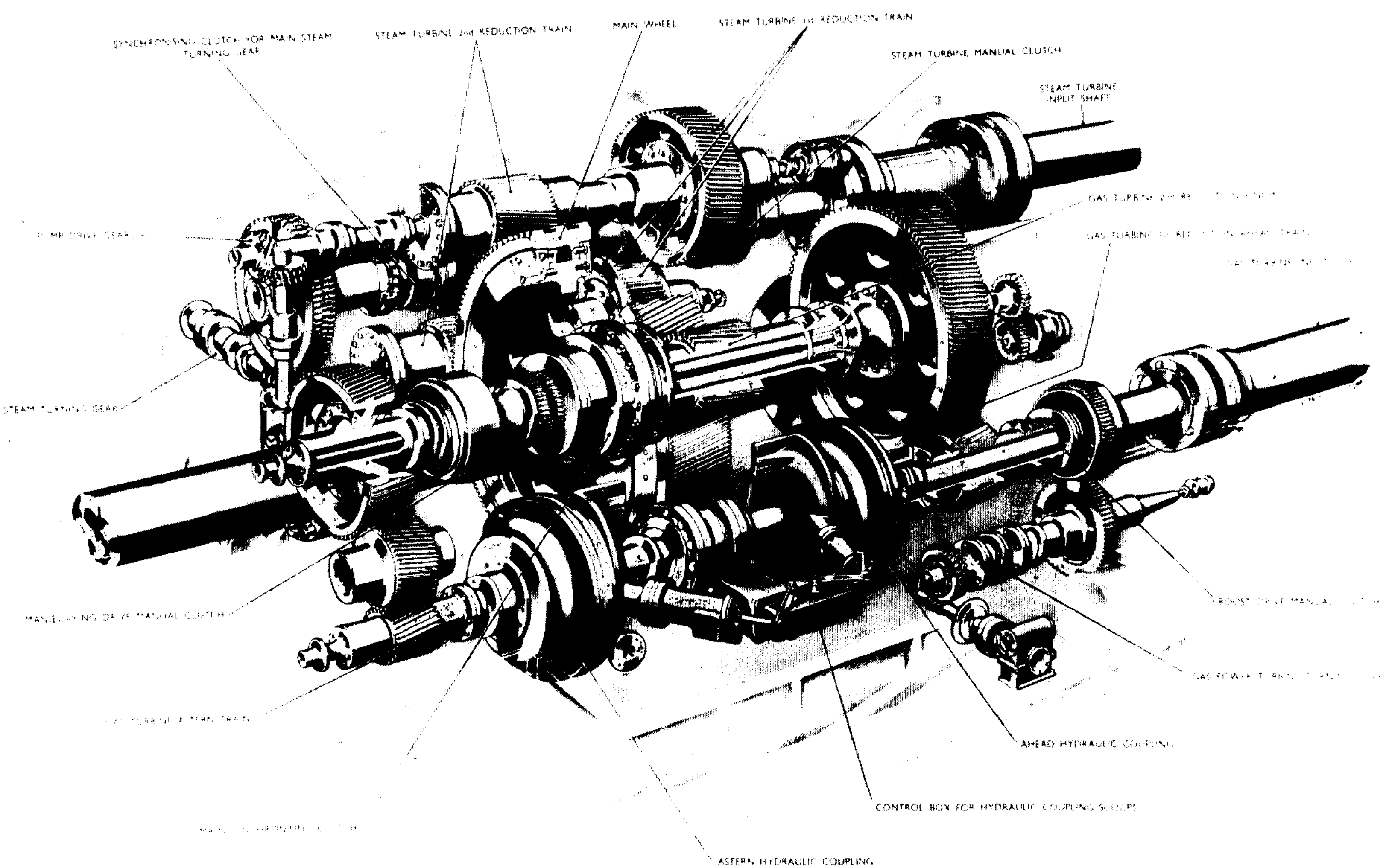


FIG. 21—Y.111A—SECTION VIEW OF GEARBOX

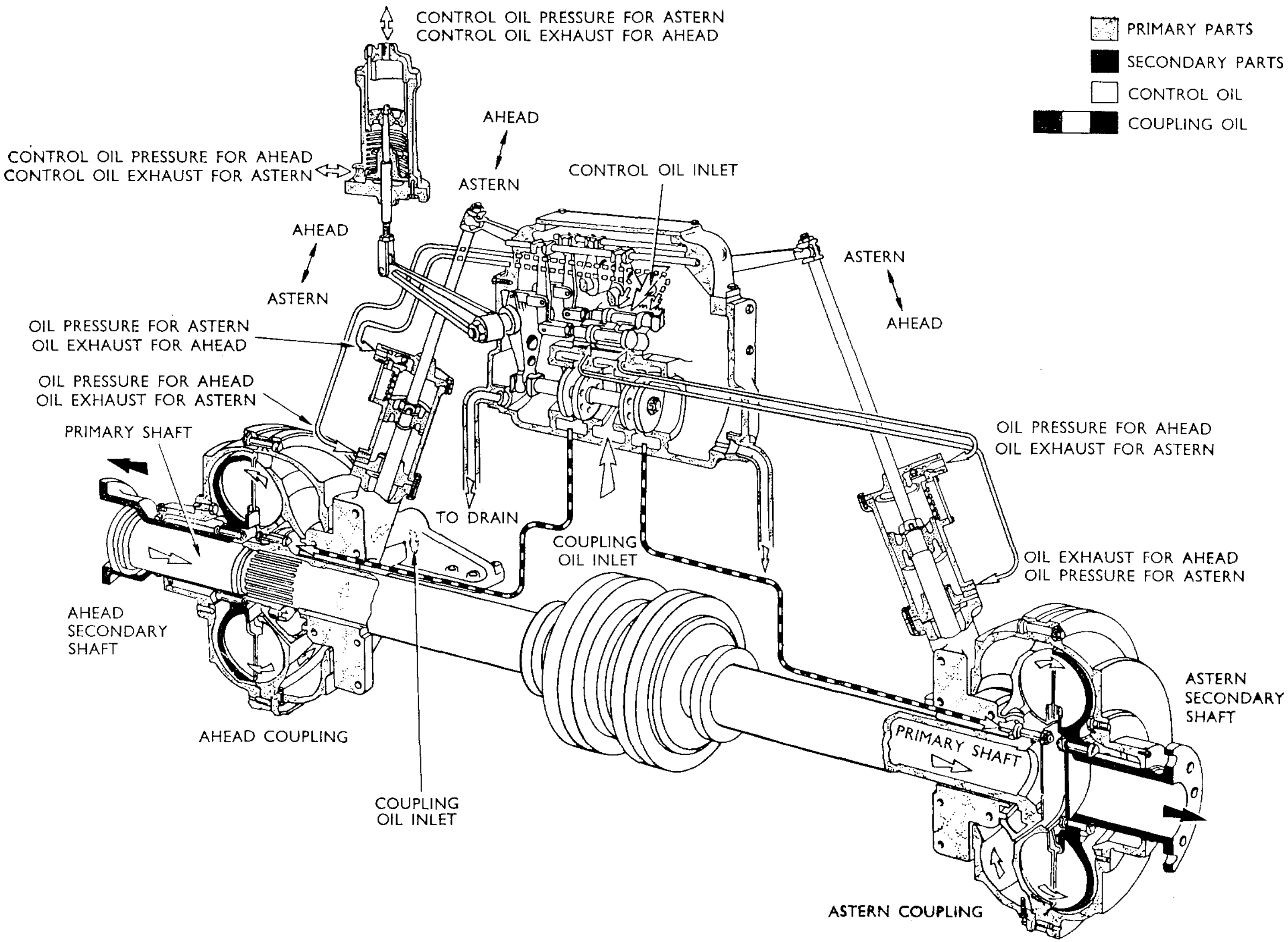


FIG. 22—DIAGRAMMATIC ARRANGEMENT OF TANDEM SET FLUID COUPLINGS WITH HYDRAULIC SERVO CONTROL

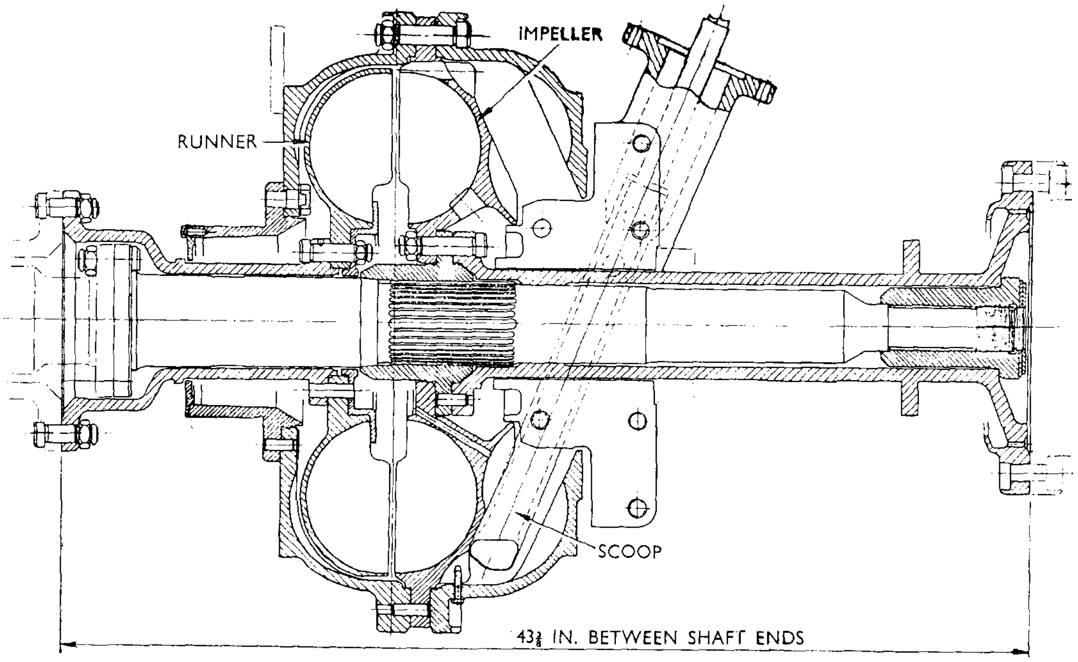


FIG. 23—AHEAD HYDRAULIC COUPLING—SECTIONAL ARRANGEMENT

ahead coupling is shown in FIG. 23. The couplings are designed so that oil supplied from the main lubricating oil system is continually flowing through the working circuit via ports in the impellers into the rotating scoop chambers where it is picked up by the scoop tubes and returned to the sump. Thus the amount of oil in the working circuit is determined by the radial position of the tip of the scoop tube. An hydraulic cylinder connected directly to the top of the scoop tube determines its radial position. The valve gear in the control unit controls the admission of oil to the two hydraulic cylinders (ahead and astern). A single hydraulically-operated double-acting cylinder controls, via links and levers, the scoop positioning valves and also the oil diverter valve which diverts oil from one coupling to the other. When running in the boost condition or on steam turbines alone the main flow of oil to the couplings is shut off but a bypass orifice provides a reduced oil flow for cooling, which escapes through leak-off orifices in the peripheries of the couplings.

The hydraulically-operated double acting cylinder has three positions : ahead, neutral and astern which give, respectively :

(*a*) Ahead scoop withdrawn and ahead coupling filled with oil. Astern scoop extended, coupling empty and rotates at 200 per cent slip less the slip in the ahead coupling.

(*b*) Both scoops partially withdrawn and the working circuits of both couplings partially filled. Both couplings rotate at 100 per cent slip. Diverter valve admits reduced flow of oil to both couplings.

(*c*) Astern scoop withdrawn and astern coupling filled with oil. Ahead scoop extended, coupling empty and rotates at 200 per cent slip less the slip in the astern coupling.

When manœuvring, the output from the gas turbine is restricted to 3,500 h.p. both ahead and astern, in order not to exceed the permitted maximum operating speed of the hydraulic couplings. In both classes of ship the corresponding propeller shaft r.p.m. are in excess of 50 per cent full power r.p.m. so that the ships' speeds and manœuvring capabilities are adequate. This limitation is imposed mainly by centrifugal stresses in the hydraulic couplings.

TABLE V—*Y.102A gearing. Leading design data*

	Gear	*Pitch circle diameter in.*	*No. of teeth*	*Normal pitch in.*	*Face width in.*	*Helix angle*	*Normal pressure angle*	*K factc (max mum*
	H.P. first reduction pinion	7·96	28					424
	H.P. first reduction wheel	33·54	118					
	L.P first reduction pinion	11·65	41	0·8	8·5	26 deg. 23 min.	20 deg.	312
Steam	L.P. first reduction wheel	33·54	118					
	H.P. second reduction pinion	12·83	33					419
	L.P. second reduction pinion	12·83	33	1·2	13	10 deg. 43 min.	20 deg.	464
	Main gear wheel	71.92	185					
	Primary input wheel	13·61	52	0·8	7	13 deg. 24 min.	20 deg.	379
	Primary output wheel	13·61	52					
	First reduction pinion	12·04	46	0·8	7	13 deg. 24 min.	20 deg.	314
	First reduction wheel	38·48	147					
Gas	Second reduction pinion	12·83	33	1·2	13	10 deg. 43 min.	20 deg.	424
	Astern pinion	8·12	31					819
	Astern idler	8·38	32	0·8	7	13 deg. 24 min.	20 deg.	530
	Astern wheel	25·65	98					

**Note*: Full power astern on one gas turbine (emergency condtiion)

The Main Synchronizing Clutches

The main synchronizing clutches are vital components of these boost installations, and although this paper is concerned primarily with gearing, it is believed that a short account of these clutches will be of interest.

It has already been mentioned that an automatic clutch was fitted in the Y.100 machinery installations, to connect and disconnect the cruising turbine when reducing and increasing power. This clutch was actuated by a frictional speed sensing device and could not be locked in engagement. Although it performed satisfactorily under the limited conditions of shore trials, it was unsatisfactory in service at sea.

A new design, of the synchro-self-shifting type, with positive pawl actuating means, was developed and gave satisfactory service during prolonged trials in H.M.S. *Scarborough* and *Keppel.* However, changes in the operational roles of these frigates now require the maximum endurance at higher speeds and the cruising turbines have therefore been removed from the majority of the ships.

The main synchronizing clutches fitted in the *Ashanti* (Y.111A) and *Devonshire* (Y.102A) Classes are also of the synchro-self-shifting type and the selection of the design was influenced by experience of the operating requirements obtained with the earlier Y.100 cruising turbine clutch. A description of the clutch and an account of the clutch trials carried out ashore will be found in the Appendix to this paper. Although, at the time of writing, seagoing experience is limited to approximately one year's service in the first ship and sea trials of three later ships, the performance of this clutch has been entirely satisfactory.

Gearing Design Details

The gears of the Y.102 shore trials set and in the majority of ships' sets are in En 36 steel and are carburized, hardened and ground. In some of the ships' sets, a number of induction hardened and ground gears in En 24 steel and nitrided and ground gears in En 40c have been installed. All gears are single helical. Leading design data of the gears are given in TABLES V and VI. It will be seen that, under normal ahead operating conditions the tooth loading does not exceed 465 K but that very much higher loads can be realized when manœuvring on gas turbines. In emergency, when manœuvring at full power astern on one gas turbine alone, the associated astern gear train in the Y.102A design is loaded to 819 K.

Table VI—*Y.111A gearing. Leading design data*

	Gear	Pitch circle diameter in.	No. of teeth	Normal pitch in.	Face width in.	Helix angle	Normal pressure angle	K factor (maximum)
team	First reduction pinion	8·81	32	0·8	7	22 deg. 20 min.	20 deg.	455
	First reduction wheel	28·91	105					
	Second reduction pinion	12·83	33	1·2	13	10 deg. 43 min.	20 deg.	336
	Main gear wheel	71·92	185					
ias	First reduction pinion	12·04	46	0·8	7	13 deg. 24 min.	20 deg.	331
	First reduction wheel	38·48	147					
	Second reduction pinion	12·83	33	1·2	13	10 deg. 43 min.	20 deg.	441
	Astern pinion	8·12	31	0·8	7	13 deg. 24 min.	20 deg.	602
	Astern idler	11·78	45					356
	Astern wheel	25·65	98					

Fig. 24—Y.102 outboard astern manoeuvring gear train after completion of bedding trials

The need for helix correction was examined in the initial design stage and, in addition to bending and twisting, slew of the pinion and wheel due to side thrust of the single helix was also considered. Calculations indicated that the slew effects might be corrective and, in addition, earlier experience with single helical gears had not revealed maldistribution of load from these causes. In view of this and the lack on data on the attitudes of journals when slewed in their bearings and of the fact that adjustable bearings were to be fitted, it was decided not to apply helix corrections.

Gear Proving Trials

Gear proving trials were carried out during the Y.102 shore trials, at no load, half load and full load, to prove the bedding of the teeth under various loads and to determine whether or not helix corrections were necessary. The bedding obtained extended fully across the face widths of the teeth and it was concluded that the stiffness of the gearcase was adequate, that helix corrections were unnecessary and that the methods of manufacture, tolerances, alignment and inspection had been satisfactory. A point of special interest is that during these trials the outboard gas astern pinion and idler ran for one hour at 819 K. The bedding was good and the teeth remained in excellent condition. A photograph of these gears is shown in Fig. 24.

To date, the Y.102 shore trials gears have completed over 1,000 hours' running including 110 hours at full power and 33½ hours continuously at 130 per cent full torque. The gears are in excellent condition. E.P. lubricating oils have been used throughout the trials.

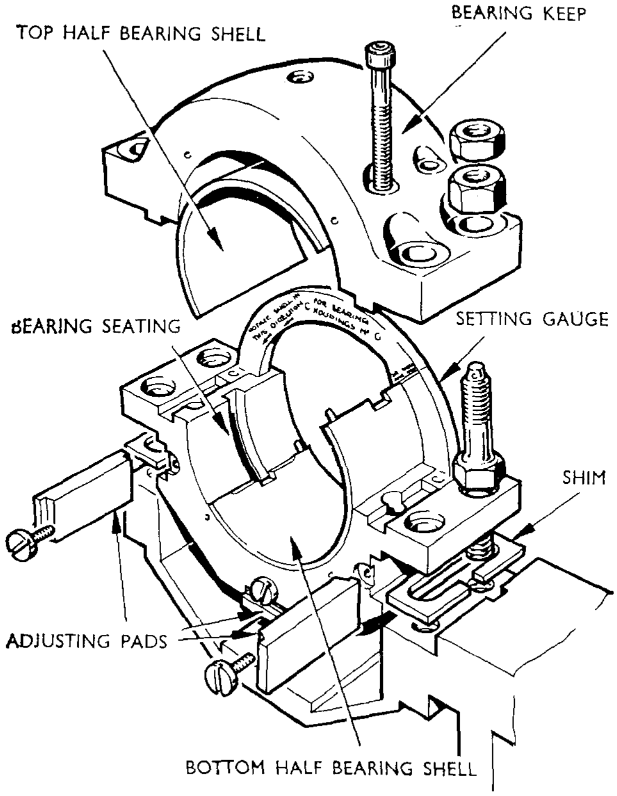


FIG. 25—TYPICAL BEARING HOUSING SHOWING SHELL AND SETTING GAUGE

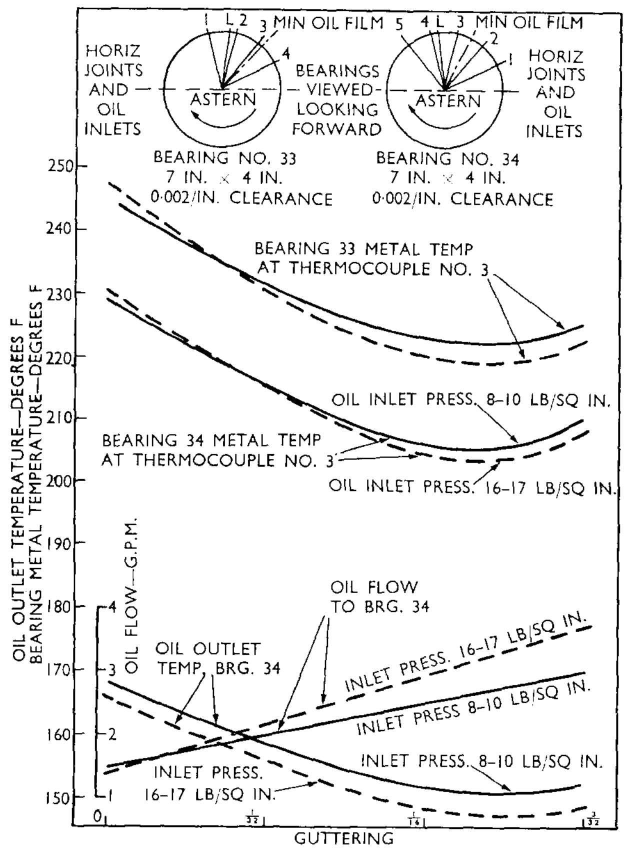


FIG. 26—EFFECT OF GUTTERING AND OIL INLET PRESSURE ON METAL TEMPERATURE AND OIL FLOW AT 3,500 H.P. AND 171 R.P.M.

Journal Bearings

The majority of the journal bearings comprise adjustable steel bearing housings into which are fitted pre-finished medium wall whitemetal lined steel shells. An interference fit is provided between shells and housings, the two parts of the latter being clamped together by socket headed bolts and studs as shown in FIG. 25. The correct radial position of the shell joint and oil inlets is achieved with a setting plate and maintained by a dowel in the bearing keep registering with a counterbore in the shell. Adjusting pads are fitted and permit true alignment during manufacture. Thermocouples located as near as possible to the positions of minimum oil film thickness are mounted radially in the bearing keeps and are embedded in small whitemetal plugs, held in contact with the bearing shells by means of screwed sleeves and springs.

With the exception of the main gear wheel bearings, all journal bearings are provided with two diametrically opposed oil inlet holes at the shell joints. During the early shore trials a number of high-speed bearings wiped or ran at high temperatures. After repositioning these bearings to provide a greater arc between oil inlet and load line, and, in some cases, increasing the diametral clearances from 0·0015 in. to 0·00225 in. per inch diameter, further troubles were avoided.

A useful expedient was resorted to on a number of occasions in order to reduce high bearing temperatures, when time did not permit bearings with larger clearances to be obtained. This involved guttering the joints at 45 degrees, to a depth of approximately $\frac{1}{16}$ in., the gutters running axially from the oil inlets to the ends of the bearing. The

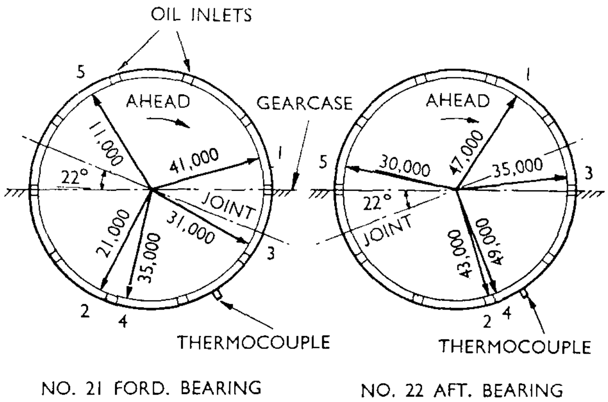


FIG. 27—Y.111A—MAIN WHEEL BEARING SHOWING LOAD LINES
(1) STEAM AHEAD (2) STEAM ASTERN
(3) STEAM AND GAS AHEAD (4) GAS AHEAD
(5) GAS ASTERN

effect is not only to increase the oil flow, as would be expected, but also to reduce the whitemetal temperature. To evaluate this practice, trials were carried out on the highly loaded, high speed gas astern idler gear bearings without guttering and with gutters $\frac{1}{16}$ in. × 45 degrees and $\frac{3}{32}$ in. × 45 degrees. The results are shown in FIG. 26 and it is seen that the optimum amount of guttering in this case was $\frac{1}{16}$ in. and, with an oil inlet pressure of 8-10 lb/sq in. the whitemetal temperature was reduced 20 degrees F. approximately, while the oil flow was increased from 1·5 to 2·5 gal/min.

Main Gear Wheel Bearings

In a boost installation of this type the various possible running conditions both ahead and astern, result in a multiplicity of load lines (FIG. 27) in the main gear wheel journal bearings, such that it is difficult to obtain a safe location for a single oil inlet.

Trials of a conventional bearing (19 in. diameter × $12\frac{3}{4}$ in. long, 0·02 in. diametral clearance and single oil inlet 2 in. × $1\frac{1}{2}$ in. with spreader) were carried out at 124 r.p.m. and oil inlet pressure 10 lb/sq in., the bearing being loaded to 232 lb/sq in. The direction of loading was varied and it was found that the bearing wiped when the load line approached within 40 degrees (approximately) of the oil inlet, in both the leading and trailing conditions.

It was therefore decided to test a bearing provided with a central circumferential groove, 1 in. wide × $\frac{1}{4}$in. deep, with twelve equally spaced oil inlet holes each 1 in. in diameter. The bearing was loaded to 345 lb/sq in. with oil inlet pressure 10 lb/sq in. and at this loading performed satisfactorily throughout the speed range 20-184 r.p.m. A further test involved four hours' running at $1\frac{1}{2}$ r.p.m. at 50 lb/sq in. loading. Evidence of slight rubbing was found afterwards but the bearing was undamaged. This type of bearing, with a circumferential oil groove, was subsequently fitted in the ships' sets. Experience so far has been entirely satisfactory.

Sea Trials Experience in H.M. Ships 'Ashanti', 'Nubian' and 'Devonshire'

Good and Dunlop[13] have given an account of the extensive sea trials of the first three ships and have described the techniques of operating the gas turbines, clutches and hydraulic coupling controls when boosting and when manœuvring on gas turbines. In this paper, no attempt will be made to describe the manœuvring trials in greater detail or to analyse the large number of records obtained. A separate paper would be required to cover this subject and it is thought that such a paper, given by the designers of this machinery, would be of considerable interest.

In all ships the performance of the gears themselves was very satisfactory, the tooth flanks generally showing good bedding and being in excellent condition after the trials. The starboard gear set in H.M.S. *Devonshire* contained a number of induction hardened gears, including the main gear wheel and these also were in excellent condition.

In the author's opinion the sea trials of these ships have shown that the gears,

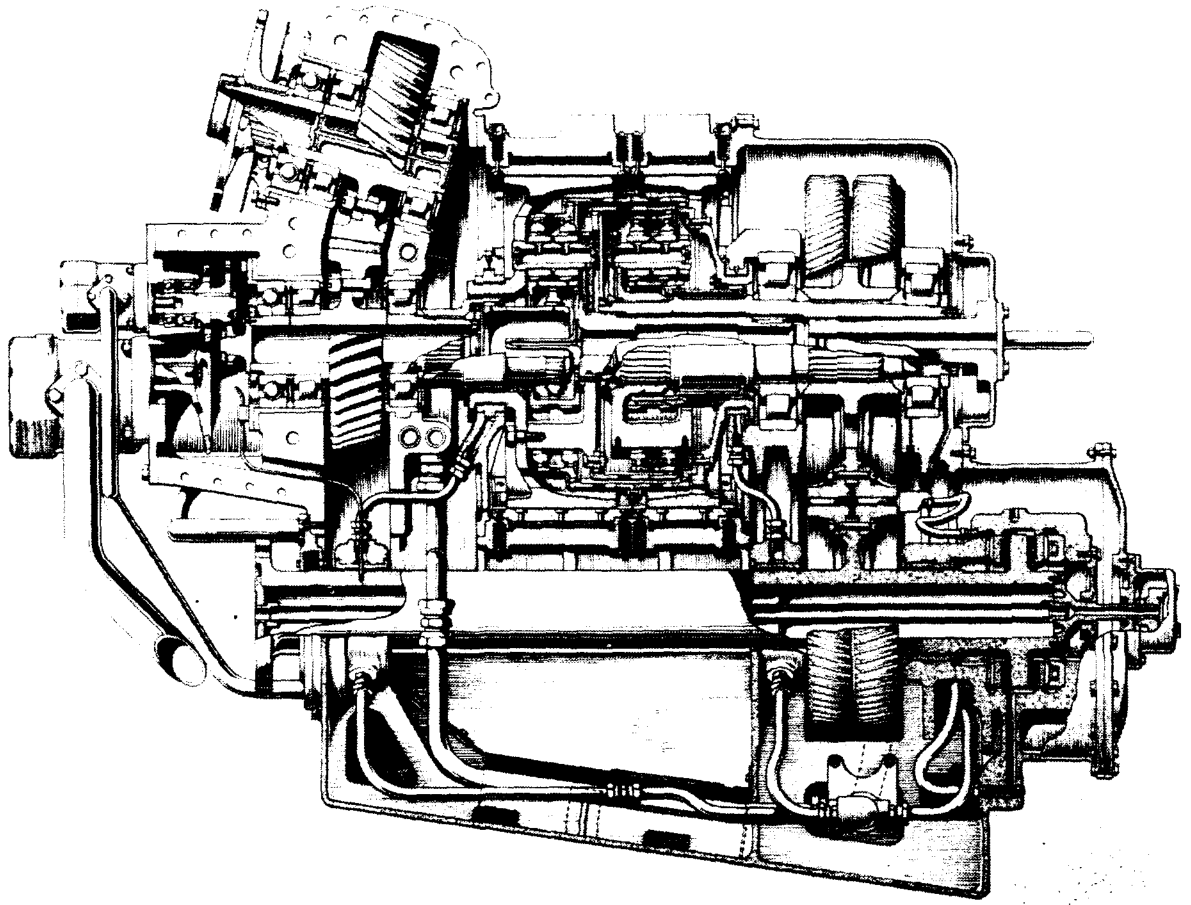


FIG. 28—BRAVE CLASS FAST PATROL BOATS' REVERSE REDUCTION GEAR

including the hydraulic couplings and main synchronizing clutches, fulfil the design requirements and, notwithstanding their complexity, will prove to be robust and durable. In stating this opinion, it is, of course, assumed that the specified standards of manufacture will, in all cases, be achieved and that the standards of operation and maintenance will be suited to the precision of the machinery.

' Brave ' Class—Epicyclic Gears

Although many epicyclic reduction gears are giving satisfactory service in turbo-generators and other auxiliary machinery in the post-war Fleet, they have so far been used for main propulsion gears only in a limited number of smaller vessels. Of these, the most interesting application has been in the *Brave* Class fast patrol boats, in which use is made of epicyclic gearing not only to give a primary reduction between the marine Proteus gas turbines and flexible cardan shafts, but also in the reverse/reduction gearboxes which incorporate carburized, hardened and ground spiral bevel vee-drive gears. In this type of warship the use of small, lightweight high powered machinery is of paramount importance and the epicyclic gear has proved to be extremely suitable for this application.

FIG. 28 shows a sectional arrangement of the reverse/reduction gear, which comprises three basic units ; the vee-drive bevel gear unit, the reverse/reduction gear consisting of two epicyclic gears in series and a parallel shaft gear unit. The construction of the epicyclic units follows the well known Stoeckicht principles and has already been described by Allen and Jones[15]. Nitrided sun wheel and planet wheels are fitted, the latter revolving on whitemetal faced spindles. The gear is capable of transmitting 3,850 h.p. through a speed reduction of 4,988-1,707 r.p.m. and is designed to transmit full power ahead or astern, thus permitting handed propellers. The total weight of the gearbox is less than two tons and the overall length is 6 ft.

Ahead or astern operation is achieved by holding one or other of the two brakes, which comprise hydraulically actuated steel and sintered bronze shoes bearing on spheroidal graphite cast iron drums. In the neutral position both brakes are freed. Before installation in the first boat, the gears were run up to full speed, in both directions, ashore but unfortunately it was not possible to carry out any running under load. It is not, perhaps, surprising that the early sea trials at full power resulted in two failures of the planet spindle bearings in the first train of the port gearbox. In each case the damage was confined to the gear train in which the failure occurred. The trouble was cured by the addition of a separate, motor driven lubricating oil pump operated by a pressure switch to supplement the supply of lubricating oil at low speeds and by the provision of slightly greater oil clearance and an additional oil supply hole in the planet spindle bearings.

Further sea trials were entirely satisfactory and fully demonstrated, not only the flexibility of the machinery installation as a whole but also the robustness of the gears. For example, the boats have been accelerated from rest to full power in 30 seconds, then decelerated to ' stopped in the water ' in the same interval. On one occasion when proceeding on three engines at more than 30 knots, one of the gear levers was inadvertently moved from the ahead to the astern position. The gearing and boat were heard to protest and the mistake was rectified. No troubles were experienced with this gearbox, which remained in service. During the course of a subsequent routine examination the only evidence that this incident had occurred was the torn appearance of the astern brake shoes, and particles of sintered bronze which had become welded to the brake drums.

In service the performance of the epicyclic units in these gear sets has been excellent and that of the gear sets as a whole has been good, particularly so when it is recalled that these were advanced, highly loaded designs and no testing or development was possible before installation in the boats. Minor troubles which have been experienced include :

(*i*) Porosity of some of the spheroidal graphite cast iron brake drums. Although no failures have occurred a number of drums have been replaced in forged steel.

(*ii*) Wear of lubricating pump-drive gears. Gears with increased face width have been fitted.

One major breakdown has occurred, caused, it is believed, by failure of a light alloy cage in a vee-drive roller bearing, permitting gross misalignment of the vee-drive bevel gears and resulting, nine months later, in fracture of the heavily pitted teeth. Steel cages have now been fitted to these bearings.

PART II

GENERAL INFORMATION

Trials of E.P. Lubricating Oils

It has already been mentioned that minor scuffing of the secondary gears occurred during the sea trials of some of the *Whitby* and *Blackwood* Class vessels and because of this it was decided that E.P. lubricating oils should be used in these ships.

At this time the Admiralty had no ship experience of such oils and it was not known whether or not the oils supplied by the various manufacturers would be compatible when mixed together. In addition to the need for such oils in the ships mentioned, it was believed that they would also be required in future ships, to enable full advantage to be taken of the higher loadings permitted by hardened and ground gears.

A large scale experiment was therefore started, in which ships of the *Whitby* and *Blackwood* Class with Y.100 gears were run on nine formulations of E.P. oils supplied by seven manufacturers. Notwithstanding the logistic problems, the various supplies of oil were not mixed, i.e. each ship used oil of one formulation only, supplied by one manufacturer.

Initially it was hoped that the use of E.P. oils would permit these gears to ' run in ' and would then prove unnecessary. This hope has not been realized, for during the course of the experiment certain ships were unable to obtain the necessary supplies and were obliged to recharge their lubricating oil systems with the conventional turbine gear oil (OM100). In some cases but not all, this resulted in further scuffing, which was once again arrested by the use of E.P. oils as soon as supplies were received.

The trials were continued over a period of five years and although only one case of scuffing occurred with an E.P. oil, various other troubles were experienced, resulting in the withdrawal from service of certain oils, and revision of the Admiralty specification which has, in some respects, been made more severe. Some of these troubles were probably not related to the presence of extreme pressure additives and may well have occurred had the conventional turbine gear oil (OM100) been used in these ships. Experience so far suggests the following conclusions :—

(*i*) The use of copper, zinc or alloys containing these metals in lubricating oil systems is undesirable, particularly where high temperatures exist, since these materials can react with the E.P. agent in the oil and the reaction products may give rise to corrosion and/or sludge formation. Other reactive metals, such as cadmium may also be undesirable and should be avoided. The use of steel pipes, valves and bearing shells is therefore recommended but the retention of non-ferrous materials for lubricating oil coolers is permissible. In particular, the use of zinc in galvanized lubricating oil filter cages and of copper in lubricating oil heater tubes should be avoided.

(*ii*) The design of lubricating oil heaters should ensure that local overheating of the oil does not occur, even when flow is restricted by partial fouling. Operating personnel must be made aware of the possibilities of overheating which can result from leaking steam valves and maloperation.

(*iii*) In machinery installations using steam at high temperatures it is most desirable that a supply of lubricating oil should be maintained, for cooling purposes, to auxiliary machines after they have been stopped. The steam temperature in the Y.100 frigates is only 850 degrees F. but apparent whitemetal temperatures of 400 degrees F. have been recorded, due to heat soakage after shutting down, in the bearings of some of the larger auxiliaries, not fitted with independently driven lubricating oil pumps. Such temperatures are suspected to have resulted in breakdown of the lubricating oil, blackening and corrosion of bearings.

(*iv*) In some steam driven auxiliaries the ingress of water to the lubricating oil was excessive and provision for removing water was inadequate. This is believed to have caused a number of bearing troubles and cases of excessive wear of gears. In one ship, hydrolysis of the E.P. lubricating oil occurred, resulting in the formation of strong acids and corrosion fatigue failure of two turbo-generator gear pinions. The oil concerned was withdrawn. For future ships, consideration is being given to the provision of an auxiliary lubricating oil renovating system.

(*v*) It was found that the lubricating oil ' make-up ' rates in these ships was sufficiently high to maintain adequate load carrying capacity of the oils and only slight reduction in the latter, due to depletion of the additives, was experienced.

(*vi*) The use of certain preservatives in gearcases and other parts which are subsequently in contact with E.P. lubricating oils can result in the lowering of the demulsification properties of the oils. It is necessary therefore to ensure by efficient flushing and recharging that the preservative is removed before operating the machinery.

Throughout these trials the Admiralty received valuable advice and assistance from members of the Admiralty Fuels and Lubricants Advisory Committee, on which are represented the major oil companies, leading marine gear manufacturers and the Admiralty.

As a result of the trials, satisfactory formulations of E.P. turbine gear oils are now available and are in service in a large number of the post-war ships the Fleet.

Lubricating Oil Filtration

Prior to the *Daring* Class, lubricating oil filtration was provided by wire mesh gauze, muslin or the self-cleaning disc and scraper type filter. In addition, centrifugal separators were fitted, operating in by-pass circuits and although intended primarily for the removal of water, they also removed quantities of solid matter. Nevertheless, in ships so fitted it was considered necessary to take bridge gauge readings periodically and the renewal of worn bearings was a laborious and expensive procedure.

To improve bearing life, particularly in double-reduction gears with their increased numbers of bearings and to permit higher bearing loadings, a finer degree of filtration was desirable. After comprehensive tests at the Admiralty Engineering Laboratory, West Drayton, it was decided to fit felt filters in the *Daring* Class ships, capable of filtering out solids larger than about 30 μ and similar filters have been used in all subsequent designs up to the present day.

After the lubricating oil system has been flushed out and the first service change of filters has been made, it has been found that, under normal conditions, a life of approximately two years can be expected from the felt elements. Usually differential pressure gauges are fitted across these filters, to give an indication that the felt element is becoming choked. Automatic by-pass arrangements are now being fitted in the latest classes.

Service experience in the *Daring* Class and subsequent ships has been satisfactory and, after the initial dirt has been removed from the lubricating oil system, bearing wear has been negligible. So far, there has been little justification for the use of filters giving a finer degree of filtration and the additional size and weight of such filters has prevented their use.

Since felt filter elements cannot be cleaned, the performance of fine gauze filters and the possibility of cleaning them has been investigated. Such gauzes showed a similar efficiency of filtration but the extended life obtained by reflux cleaning has not warranted the added complication and weight of pipes and change-over valves involved. In addition, compared with felt, gauze elements are more expensive and, when choked with dirt, have shown a tendency to disintegrate.

In addition to felt filters, it is still customary to fit centrifugal purifiers and, resulting from tests carried out at the Admiralty Engineering Laboratory, it has been found possible to increase the throughput of oil from three to four times that permitted without significant decrease in performance.

Cleaning and Flushing

A problem that is still receiving serious attention is the matter of cleanliness. Scoring of teeth has occurred in a number of Y.100 gearboxes and far too many bearing shells have had to be renewed after shop trials in Y.100, Y.102A and Y.111A gearboxes for the same reason.

The scoring has been partly caused by mill scale and rust from the fabricated ferrous material used in the gearcase and lubricating oil system and partly from machining and other debris left in the gearcase and system. Pickling followed by phosphating is now specified for the removal of mill scale and rust on fabricated ferrous material but the removal of the debris left in oilways, housings, pipes, etc., is a much more difficult matter, particularly when the gearbox has been assembled. It must therefore be tacked during the final erection and with this in view revised instructions are being issued. The basis of these instructions is that all machining and fitting work must be completed before final erection and that all components must be clean, scale free and suitably protected to prevent contamination or corrosion during assembly. Personnel should be impressed with the need to maintain scrupulous cleanliness and, among the psychological aids to this, is the provision of transparent plastic sheeting to cover opened machinery, clean overalls, rubber soled shoes, vacuum cleaning facilities and adequate lighting. The erection site should be clean and where possible, covered in by a plastic tent.

After final assembly and before any running in is commenced, the lubricating oil systems must be flushed with a copious supply of clean, filtered hot oil (120—170 degrees F.) preferably section by section in order to obtain the maximum flow of oil. Flushing may take anything up to 48 hours.

When machinery is first run on shop trials it is to be run in the astern direction to minimize the danger of damage to the ahead bearing surfaces. Similar precautions will be followed before machinery is run in the ship.

Journal Bearings

Newman[14] has given an account of bearing tests carried out at Pametrada for the Admiralty and this work resulted in the use of steel backed bearings with thin whitemetal linings in the Y.100 Mark I gears and in a considerable increase in bearing loadings. As a general rule, maximum loadings of 500 lb/sq in. are permitted for pinion and primary wheel journal bearings and 350 lb/sq in. for main wheel bearings. Experience at these loadings has been entirely satisfactory and the resulting reductions in bearing length (L/D ratios $\frac{1}{3}$-$\frac{2}{3}$ are permitted for all high speed bearings) have helped to reduce gearing losses. After successful experience in Diesel engines and a limited but successful trial in the Y.E.A.D. 1 gears, prefinished whitemetal lined steel shell bearings were fitted in the Y.100 Mark II gear sets and all subsequent designs. Prefinished bearings have the following advantages over the conventional type of thick shell bearing.—

(*a*) Interchangeability of shells of the same nominal bore size
(*b*) Spares are lighter and more easily stored
(*c*) Cheaper. The cost can be as low as 10 per cent of the conventional type
(*d*) No re-metalling problems.

Above all, the use of this type of bearing provides an attractive solution to the problem of obtaining and maintaining gear alignment, a matter of particular importance with higher gear loadings and where silent running is required. With conventional bearings the use of boring jigs has been successful in some cases but the high cost of producing and maintaining such jigs is unattractive for small, peace-time building programmes. The alternative of using large, super accurate jig type boring machines[16] also permits the desired standards to be achieved but involves high capital costs for such machines and seems unlikely to be adopted by many firms.

With prefinished shell bearings manufactured to close tolerances on bore and outside diameter, it is only necessary to provide a means of adjustment between the bearing and the supporting structure. Subsequent bearing renewal can then be effected within the alignment tolerances without any fitting woık. This has been of particular advantage in several cases, where bearings have been damaged, during shop trials, by dirt and metal swarf.

Two types of prefinished bearing shell are now available:

(*a*) Medium walled and
(*b*) Thin walled,

comparative figures for a typical bearing of 7 in. bore being :

	Thin	*Medium*
Wall thickness ..	0·1425 in.	0·2188 in.
Whitemetal thickness	0·02 in.	0·02 in. nominal
Weight	2·8 lb	4·3 lb
' Nip '	0·005 in.	0·0122 in
Clamping load ..	10,000 lb	13,000 lb.

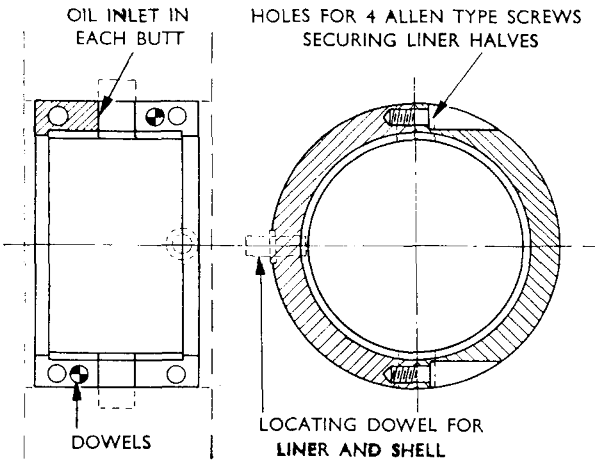


FIG. 29—Y.100 MK. II—ARRANGEMENT OF JOURNAL BEARING

In the Y.100 Mark II design, the original housing bore sizes were retained in case it became necessary to change back to the conventional bearings fitted in the Mark I design. Split liners or sleeves were used to make up the differences in thickness and to clamp the medium wall bearing shells. The sleeves had a slight interference fit in the housings and, where necessary, they were bored eccentric to correct any misalignment of the housing bores. This design has been satisfactory but it would have been preferable to use thicker sleeves to provide the necessary interference or ' nip ' of the medium walled shells. The general arrangement of this design is shown in FIG. 29. Lubricating oil is supplied to the bearing via a circumferential groove in the sleeves and oil inlets in the butts of the shells. The sleeve type of bearing has been used in a later design with thin walled bearings. With these it is not possible to provide an oil groove in the bore of the sleeve, as the shells require support over their full length and the groove is therefore machined around the outer diameter of the sleeve. The adjustable type of bearing used in the Y.102A and Y.111A designs has already been described (FIG. 25). A further advantage of this type of bearing has been the ability it confers to adjust the gear alignment, if required, after first running on load. In a number of ships it has been found that although the gear meshing was apparently satisfactory during shop trials at light load, examination after sea trials has shown that adjustment was desirable. A problem arising with prefinished bearings concerns the question of undersize journals which have been machined in error or to remove damage. With the conventional bearing it is possible to re-metal and bore undersize to suit the journal but with prefinished bearings it is necessary to adopt a standard range of undersizes. It is usually desirable to limit the whitemetal thickness to avoid too much reduction in fatigue strength and, at present, the Admiralty has made provision for standard undersizes of 0·02 in. and 0·04 in. on diameter. It is also possible to increase the thickness of the steel backing of the shell but care is necessary to ensure that the clamping bolts are capable of providing the increased ' nip ' which is required. Many cases of small errors in journal size have been rectified by nickel plating, provided the finished thickness does not exceed 0.012 in.

TABLE VII—*A.V.G.R.A. first reduction gear tests.—Nitrided En 40c wheel. All testing carried out at 6,000 r.p.m. pinion speed*

Duration hrs.	*Horse-power*	*Loading*		*Cycles* $\times 10^6$	
		K value	*lb/in. face tangential*	*Pinion*	*Wheel*
36 *hrs. approximately*	*Running up to* 8,250	0—398	0—2,652		
30	9,000	434	2,890	10·8	2·47
30	9,750	470	3,130	10·8	2·47
30	10,500	506	3,370	10·8	2·47
30	11,250	542	3,610	10·8	2·47
30	12,000	578	3,850	10·8	2·47
32	12,750	614	4,090	11·5	2·63
29¾	13,500	651	4,340	10·7	2·44
28½	14,250	687	4,580	10·3	2·35
30	15,000	723	4,810	10·8	2·47
30	15,750	759	5,050	10·8	2·47
29½	16,500	795	5,300	10·6	2·42
30	17,250	832	5,545	10·8	2·47
25¼	18,000	868	5,780	9·1	2·08
Torque reversed to load other wheel flank and a 10^8 wheel cycle run commenced.					
4	—	289—1,060	—	1·44	0·33
259	22,000	1,060	7,060	93·3	21·35
Pinion tooth cracked. Wheel and direction of loading reversed so that damaged astern pinion flank used to load the original ahead wheel flank and test continued.					
¼	22,000	1,060	7,060	0·1	0·02
Cracked tooth broken off pinion. Wheel undamaged. New pinion installed.					
613¾	18,000	867	5,780	220·8	50·5
288¼	22,000	1,060	7,060	104	23·75
Testing stopped for replacement of loading wheel. Test gears undamaged.					
59¼	22,000	1,060	7,060	21·3	4·87
30	24,000	1,156	7,700	10·8	2·47
30	26,000	1,254	8,360	10·8	2·47
30	28,000	1,350	9,000	10·8	2·47

Test gears undamaged at end of these tests.

Recent Full-Scale Gear Tests

Newman[6] has described some of the full-scale gear tests carried out by A.V.G.R.A. and which have played an essential part in the development of the surface hardened and ground gears now in service. Further tests have been carried out in recent years and are briefly described below. Since descriptions of the two test rigs have already been published[6] they will not be repeated here.

The particulars of the test gears are as follows :

	First Reduction Test Rig		*Second Reduction Test Rig*	
	Pinion	*Wheel*	*Pinion*	*Wheel*
Pitch circle diameter, in.	8·186	35·814	13·256	70·208
No. of teeth	24	105	27	143
Normal pitch, in.	1·047 (3D.P.)		1·496 (2D.P.)	
Normal pressure angle ..	20 deg.		16 deg. 35 min.	
Helix angle	12 deg. 14 min.		14 deg. 07 min.	
Face width, in.	8		10	
Speed (r.p.m.)	6,000	1,370	1,500	283

First Reduction Gear Tests—Nitrided Gears

Chamberlain[4] and Newman[6] described the manufacture and initial tests of an En 40c nitrided wheel and further testing was carried out after the publication of their papers. Details are given in TABLE VII from which it will be seen that after 21 $\times$ 10^6 wheel cycles at 1,060 K a pinion tooth was found cracked. In an attempt to prolong the test, the nitrided wheel rim and direction of loading were reversed but the cracked tooth failed immediately the tests were recommenced.

A new carburized and ground pinion was installed, 50 $\times$ 10^6 wheel cycles at 867 K were successfully completed and the load had been raised to 1,060 K when the cast steel centre of the loading wheel collapsed and caused fracture of the loading wheel rim. The nitrided test wheel remained in excellent condition.

After replacement of the loading wheel, tests were resumed and the nitrided wheel successfully withstood over 2 $\times$ 10^6 cycles at each of the loads corresponding to 1,156 K, 1,254 K and 1,350 K.

The carburized and ground pinion was then replaced by an En 40c nitrided and ground pinion. Testing was continued on the astern wheel flanks II and 3·6 $\times$ 10^6 pinion cycles were completed at each of the following loads : 578 K, 675 K, 770 K, 868 K, 963 K, 1,060 K, both pinion and wheel remaining in satisfactory condition. Testing was then resumed on the ahead wheel flanks I and the following programme was satisfactorily completed : 3·6 $\times$ 10^6 pinion cycles at each load, equivalent to 578 K, 723 K, 868 K, 5 $\times$ 10^6 cycles at 1,060 K and 1·08 $\times$ 10^7 cycles at 1,156 K. After 0·96 $\times$ 10^7 cycles at 1,254 K a pinion tooth broke and, in passing through the mesh, caused severe damage to the remaining teeth and to the wheel. Metallurgical examinations are not yet complete but it is known that the depth of nitrided case of this pinion did not exceed 0·015 in.

Second Reduction Gear Tests

(*i*) *En 24 Induction Hardened and Ground Wheel* : Testing of a 2 D.P. En 24 induction hardened wheel was also described by Newman[6] and further tests on this wheel were carried out after the publication of his paper. The back-tempered wheel flanks, running against a carburized hardened and ground En 36A pinion, were run in, before increasing the load to 909 K. After 5 $\times$ 10^6 wheel cycles at this loading, pitting again began towards one end of the wheel teeth and increased somewhat throughout 50 $\times$ 10^6 wheel cycles. The load was then increased to 1,360 K and, after 1·3 $\times$ 10^6 wheel cycles the teeth scuffed over the full face of the wheel and the test was abandoned. Subsequent tests of the lubricating oil (OEP.90) showed that it had suffered an appreciable reduction in load carrying capacity. Metallurgical examination showed that the position of the pits on the wheel teeth corresponded to soft areas at the ends of the teeth where hardening had been imperfectly carried out. Both the equipment and operating technique have been improved

FIG. 30—AVGRA—2ND REDUCTION EN40C NITRIDED PINION

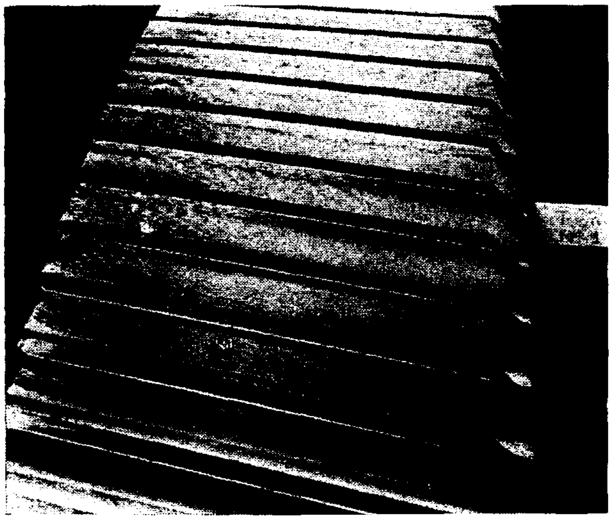
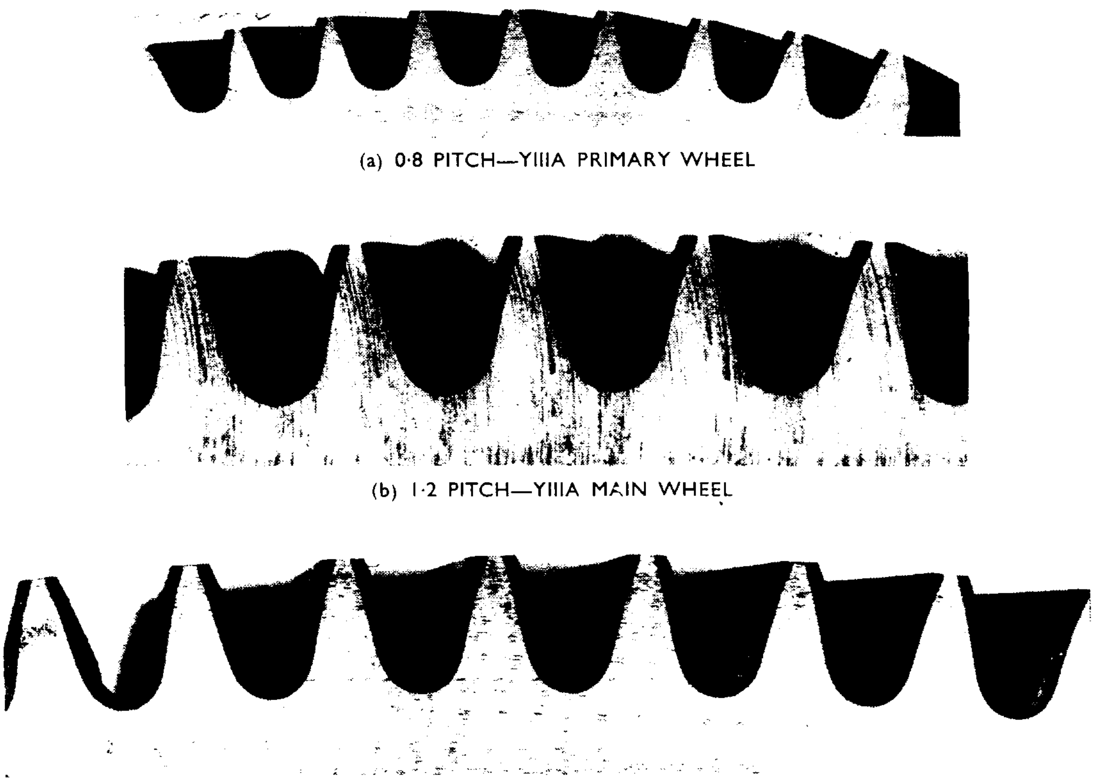


FIG. 31—AVGRA—2ND REDUCTION EN9 WHEEL

since this wheel was induction hardened and today it is possible to achieve satisfactory hardening throughout the length of the teeth.

(*ii*) *En 9 Induction Hardened and Ground Wheel* : Testing of this 2 D.P. wheel was commenced with an En 36A carburized and ground pinion. After short runs at lower powers the load was worked up to 904 K (10,146 lb/in. face) and, after $21{\cdot}7 \times 10^6$ wheel cycles tooth breakage of the pinion occurred. The wheel was undamaged and testing was resumed using an En 35 carburized and ground pinion, which had already undergone earlier tests with other wheels. After another $8{\cdot}3 \times 10^6$ wheel cycles at 904 K this pinion also failed, the wheel fortunately remaining in excellent condition.

(*iii*) *En* 40*c Nitrided Pinion* : Testing was continued against an En 40c nitrided pinion. This pinion was ground after hobbing, only to improve the surface finish and obtain the desired helix, and profile corrections. After nitriding, the pinion was checked for alignment and meshing and testing was begun. Some heavy bedding was observed during the early stages of testing and the roots of the pinion teeth were therefore honed. Subsequent bedding was excellent and this pinion had completed 10^8 load cycles ($18{\cdot}9 \times 10^6$ wheel cycles) at 904 K when repair of the loading wheel became necessary and testing had to be halted. The wheel had then completed $46{\cdot}9 \times 10^6$ wheel cycles at 904 K on the same flank and was in excellent condition, as was the underground nitrided pinion. FIGS. 30 and 31 illustrate the appearance of the pinion and wheel after 58×10^6 and 39×10^6 cycles, respectively, at 904 K. It will be noticed that the matt finish produced by the nitriding process is still evident, except in the roots, where honing was carried out. Testing was resumed after repairs to the load wheel and, at the time of writing, 10^6 wheel cycles at 1,023 K have been successfully completed. Further tests are planned, to include 10^6 wheel cycles at 1,137, 1,250 and 1,364 K.



(a) 0·8 PITCH—YIIIA PRIMARY WHEEL

(b) 1·2 PITCH—YIIIA MAIN WHEEL

(c) 1·5 PITCH—AVGRA TEST WHEEL

FIG. 32—INDUCTION HARDENED GEARS—HARDNESS CONTOURS AFTER ETCHING

Induction Hardening

Chamberlain[4] has described A.V.G.R.A.'s earlier work in this field and mentioned the installation of a new gear induction hardening machine, in which the gear is immersed in oil. Since then, further development has been carried out and this machine was used to harden, at 8·3 Kc/s, the En 9 second reduction test wheel described above and a number of production gears, including two main gear wheels and ten primary wheels, for the G.M. destroyers and G.P. frigates. Etched hardness contours for three of these gears are shown in FIG. 32 and these are typical.

So far, it has not proved possible to dispense with the 'before' and 'after' test arcs, which are used to establish and check suitable hardening conditions. A satisfactory method of measuring, non-destructively, the depth of hardening is still being sought.

It has, however, proved possible to obtain satisfactory hardness contours at the ends of the tooth faces and, by the use of quenching jets, suitably directed to reduce considerably the 'back tempering' effects. Typical hardness figures measured on the production gears mentioned above were in the range 550-650 V.P.N. on the ahead flanks and 540-640 V.P.N. on the astern flanks. The ahead flank of the first tooth, which is back tempered during hardening of the last tooth was some 20-50 V.P.N. softer than the other teeth.

Further work remains to be done to investigate the effects of variations in inductor size and clearances, alternative quenching media and to establish the minimum size of teeth which can be satisfactorily hardened at 8·3 Kc/s.

The response of various steels to induction hardening and their load carrying capacities as gears, when hardened, are also being investigated. Fifty-eight 8 in., P.C.D., 4 D.P. power circulator gears, in ten different steels, have been induction hardened and will be tested.

TABLE VIII—*Comparison of weights of post-war gear designs*

Design	*S.h.p.*	*Reduction ratio*	*Lloyd's K value*		*Weight tons*
			Primary	*Secondary*	
Daring I	27,000	17·7	83 94	100	44
Daring II	27,000	22·9	130	125	37½
Diana	27,000	23·2	236 260	200	32
Y.E.A.D. 1.	30,000	H.P. 38·9 L.P. 30·2	436 353	422	27½

Note :—Weight of thrust block included in all cases.

With fine pitch gears, shallower hardened contours are required and induction hardening at higher frequencies is necessary. Preliminary trials have been completed and a 3 ft diameter 6 D.P. first reduction test wheel is to be induction hardened, at 250 Kc/s, and tested.

CONCLUDING REMARKS

Only a few gear designs have been required for the Royal Navy's post-war shipbuilding programme and in each of these, very considerable departures from past and proved practice have been made.

Thus is the *Daring* Class destroyers, double-reduction gears were fitted for the first time in British warships and although this step forward appears to have been taken with considerable misgivings at the time, it was, nevertheless, successfully accomplished.

In the Y.100 gears, harder alloy steels were utilized at tooth loadings more than twice as high as those in the *Daring* I and II designs and substantial reductions in size and weight were achieved. Although some troubles were experienced, only one of the original twenty-four Mark I sets has been renewed and it is probable that this set would have continued to run satisfactorily for a considerable period. These troubles were entirely eliminated in the Mark II design by the use of carburized, hardened and ground secondary pinions.

Although the excellent performance of the *Daring* III (*Diana*) and R.C.N. *St. Laurent* Class gears had already demonstrated the advantages of carburized, hardened and ground gears, the Y.E.A.D. 1 design was the first British attempt to utilize such gears, and its successful performance on shore trials was encouraging and gratifying.

In all these gear designs, the desire to reduce size and weight, without sacrifice of reliability, was of primary importance. Comparisons of gear sizes and weights can be misleading, unless the gears compared are of similar type, power and reduction ratio but the relevant details of the designs mentioned above are shown in TABLE VIII and it will be seen that the three *Daring* Class and the Y.E.A.D. 1 designs are of comparable power, although the reduction ratio of the latter is considerably greater. Notwithstanding this, the use of hardened and ground gears and shorter, more highly loaded bearings has resulted in substantial reductions in size and weight without sacrifice of reliability.

The Y.102 and Y.111 gear designs are so unlike their predecessors that comparisons of weights and sizes are meaningless—but had it been necessary to use through hardened, hobbed and shaved gears in these designs, the authors believe that this would have necessitated larger machinery spaces in the ships. The experience obtained during manufacture and in the shore trials and sea trials of these gears has already pointed the way to more compact and efficient gears of similar type. Should a requirement for further boost machinery installations, or perhaps, an all-gas-turbine ship arise, it is thought that the design and manufacture of the gears could now be undertaken with confidence.

Thus, in the period under review, the transition has been made from the comparatively large but simple, single-reduction, hobbed and filed gears fitted in war-time construction, to the complex double-reduction, highly loaded, hardened and ground gears fitted in the latest warships. It can be said that A.V.G.R.A.'s original objectives have, in large measure, been attained and although much work remains to be done, this seems likely to result more in consolidation of the present gains than in further substantial reductions in weight and size of naval gears.

The development and testing of induction hardened and nitrided gears will continue and experience so far suggests that they will replace carburized gears in future designs, with substantial reductions in manufacturing costs. Highly loaded gears of both types are now going into service and tests of an induction hardened primary wheel (6 D.P.) and nitrided secondary wheels (4 D.P. and 6 D.P.) will be commenced shortly. Notwithstanding the big improvements in gear cutting accuracy which have been made during this period, it is believed that even greater accuracy will be required in future warship gears. At present, B.S. 1807 : 1952 Grade A1 standards are specified and can be attained by the majority of British warship gear manufacturers. It is doubtful whether further improvements in gear accuracy can be made with the gear measuring equipment in general use at present. The use of better equipment, now becoming available, is most desirable and should permit the present standards of accuracy to be assessed with more certainty and, it is hoped, to be surpassed in the future.

Although the use of adjustable journal bearings in recent designs permits correction of misalignment which may be apparent after full power trials, the authors believe that the problem of maintaining good alignment, throughout the wide range of powers at which naval vessels operate, requires further study. Compared with the gears themselves, the design and construction of gearcases seems to have received little attention and it may be that further work in this field, including, perhaps the use of model structures, would be rewarding. In addition to the maintenance of gear alignment there is a need for improvements in design to permit thorough cleaning during assembly. The troubles experienced from dirt and lack of care during manufacture have been referred to earlier in this paper but the resultant delays and expense are so great that no apology is made for again stressing the need for improvement.

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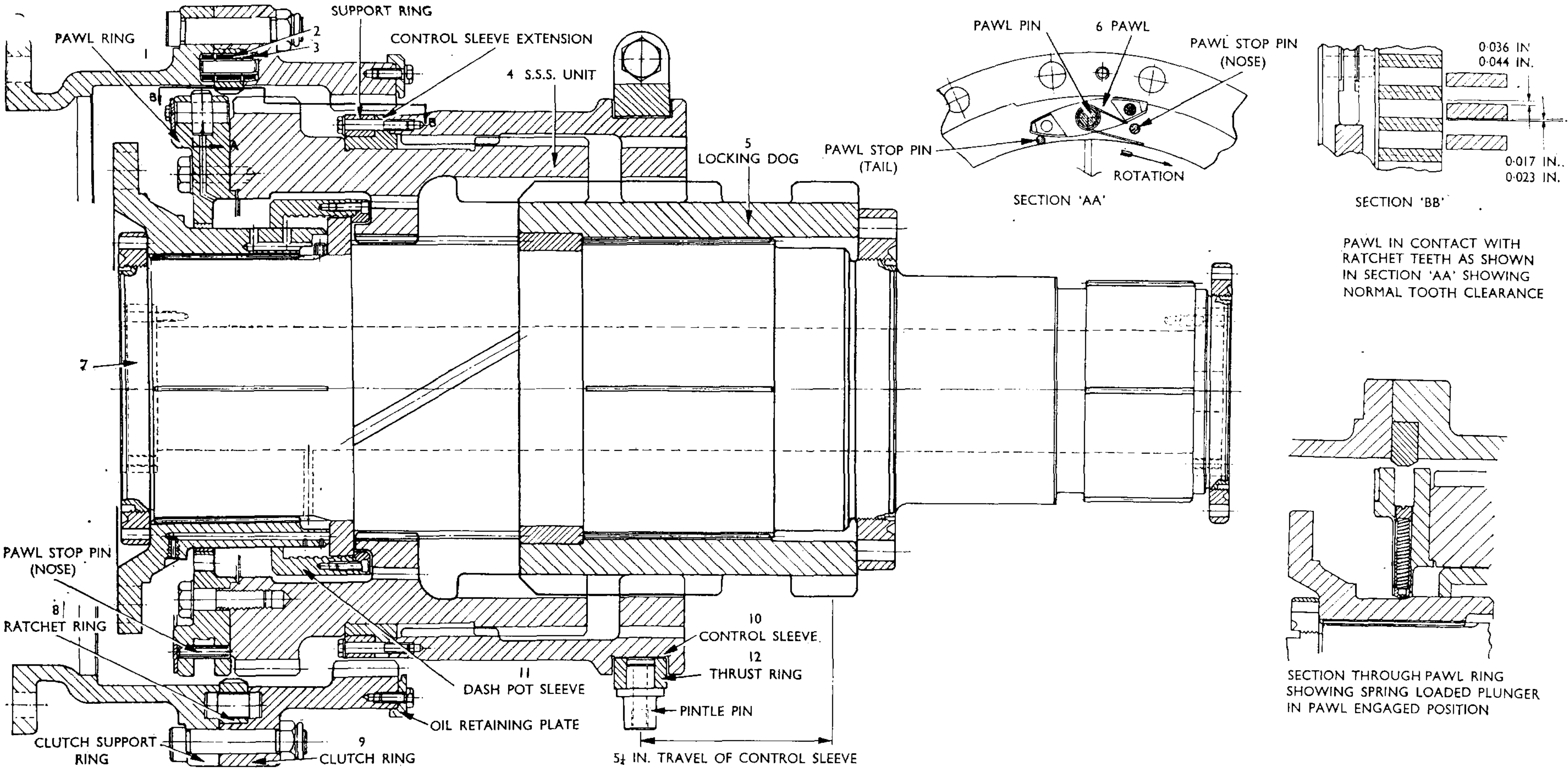

1.	Pawl ring	5.	Locking dog teeth	9.	Clutch ring
2.	Metalistic bushes	6.	Spring-loaded pawls	10.	Control sleeve
3.	Metalistic bushes	7.	Helically splined shaft	11.	Dashpot sleeve
4.	S.S.S. unit	8.	Ratchet ring	12.	Thrust ring

Fig. 33—Assembly of main synchronizing clutch—Port

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APPENDIX

THE MAIN SYNCHRONIZING CLUTCH (Y.102A AND Y.111A)

Description

A sectional arrangement of the clutch is shown in FIG. 33. The gas turbine is connected, via the intermediate gearing, to a helically splined shaft (7) on which slides the S.S.S. Unit (4). The pawl ring (1) is bolted to the S.S.S. unit and carries four pairs of case hardened, spring loaded pawls (6) which can engage with the ratchet ring (8) mounted on Metalastick bushes (2, 3) to the clutch ring (9). A control sleeve (10) is connected to the S.S.S. unit through straight splines. The positions of the various parts of the clutch during an engagement are shown diagramatically in FIG. 34.

When the associated gas turbine is out of use, the clutch is locked in the disengaged ' pawls free ' position permitting free rotation of the clutch ring in either direction, as when manœuvring on the steam turbine. Gas turbines are only started with the propeller shaft stationary or rotating ahead and movement of the gas turbine starting lever actuates relays to unlock the clutch and put it into the pawls engaged position. To do this the control sleeve is moved through a spring link mechanism by means of a double-acting hydraulic cylinder and piston. Thus when oil is transmitted to the cylinder the piston moves its complete travel ($5\frac{1}{4}$ in.) but the control sleeve only moves $3\frac{1}{4}$ in. since its teeth abut axially against the locking dog teeth (5) and the spring link is compressed

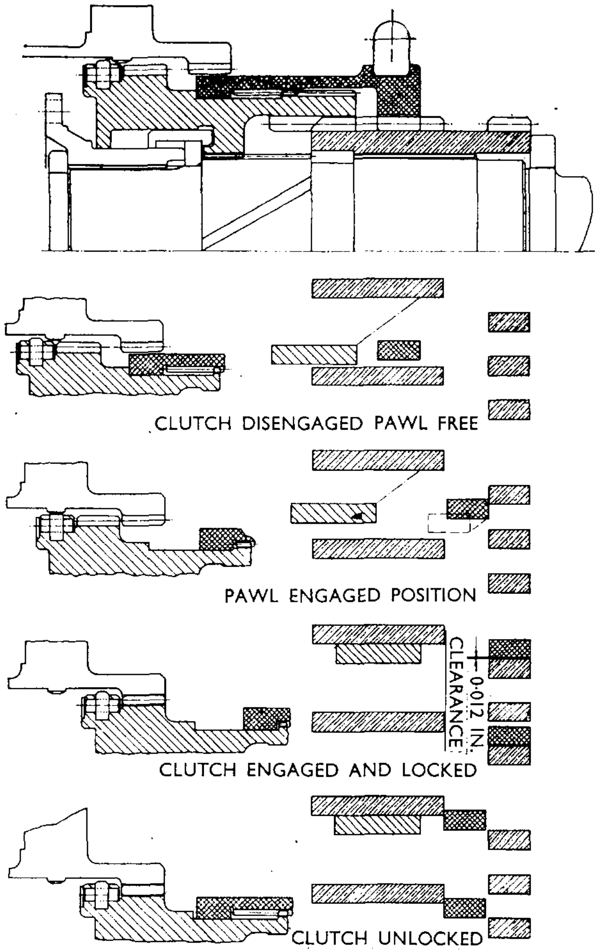

FIG. 34—Y.102A—ASSEMBLY OF MAIN SYNCHRONIZING CLUTCH—PORT. POSITIONS OF VARIOUS PARTS DURING ENGAGEMENT AND DISENGAGEMENT

2 in. to take up the difference in movement. This axial movement causes the pawls to line up in ratcheting relationship with the ratchet teeth.

When the clutch has moved to the pawls engaged position, the gas turbine is automatically started and immediately the clutch primary components tend to pass through synchronism with the clutch ring, the pawl tips meet the ratchet teeth and precisely align the teeth of the S.S.S. unit with the clutch ring tooth spaces. The pawls then transmit the small force required to initiate movement of the S.S.S. unit along the helical splines to engage the clutch driving teeth. During this travel the pawls pass axially out of engagement with the ratchet teeth so they do not transmit any driving torque. As the S.S.S. unit travels towards engagement, it abuts against the dashpot sleeve (11) for the last $1\frac{1}{4}$ in. of its travel, thus cushioning the action, as the clutch takes up the drive.

The S.S.S. unit completes its helical travel when its internal driving teeth contact the flanks of the teeth on the locking dog (5). The rotational movement of the control sleeve (10) during the helical travel of the S.S.S. unit precisely aligns the locking teeth whereupon the control sleeve completes its travel to lock the clutch. After locking, the clutch is then able to transmit the drive in both the ahead and astern directions.

Preliminary Clutch Trials

Preliminary tests were made in a special rig in order to prove the pawl mechanism and establish that it could withstand long periods of continuous ratcheting without wear. A special rig was also used to permit repeated engagements and disengagements of the clutch at operational speeds but with differential accelerations considerably higher than those expected in service. Early running experience showed the need for minor modifications and subsequently some 1,500 clutch engagements were satisfactorily carried out and the clutch was stripped and found to be in excellent condition.

Shore Trials in Y.102 Installation

During trials, in moderate seas, of the original type of Y.100 cruising turbine clutch in H.M.S. *Whitby*, cruising pinion accelerations and decelerations of the order of 300 r.p.m./sec. were measured. These arose solely from the movement of the ship in a seaway and it was apparent that with the ship under helm in rough seas this figure would have been exceeded.

In the Y.100 installation the gear reduction ratio between clutch and propeller shaft is 54·6 : 1 and thus speed variations in the latter, due to sea effects, were very considerably magnified at the clutch. In the Y.102 and Y.111 installations the corresponding ratio is 5·609, or approximately one tenth that in the Y.100. Furthermore, since the clutch runs at intermediate shaft speed, the acceleration of the incoming gas turbine is also reduced by the appropriate gear ratio (3·196) at the clutch input shaft. The estimated maximum differential acceleration across the clutch is 60 r.p.m./sec. or approximately one-ninth that of the Y.100 clutch. The Y.102 clutches were designed to engage satisfactorily under maximum differential accelerations of 100 r.p.m./sec. and were in fact, tested in the shore trials installation at 108 r.p.m./sec. To achieve this condition, a technique was evolved in which the incoming gas turbine was accelerated rapidly and at the same time the other gas turbine was quickly reduced to idling from maximum speed. By varying the rates of acceleration and deceleration and by using different settings of the Froude brake on the propeller shaft it was possible to test the clutches over the complete range of speeds and at differential accelerations up to approximately twice as high as those expected in service. A further important advantage of installing the clutch in the intermediate shaft system, as in Y.102 and Y.111 is that when the associated gas turbine is at rest the high speed gearing is stationary.

Soon after the commencement of clutch trials and after successfully carrying out a number of engagements at various speeds and small differential accelerations, it was observed that the inboard clutch failed to engage at synchronism. This clutch was stripped and it was found that only the pawls, pawl carrier and ratchet ring were severely damaged and the remaining parts of the clutch were in satisfactory condition. The circumstances and cause of the failure were not conclusively established but investigation showed that :

(*a*) Due to the experimental nature of the trials, with interlocks out of action, the clutch could have been moved to the ' pawl engaged ' position at a moment when a negative speed differential existed between the ratchet ring and the pawls, e.g. when the input was already rotating faster than the output shaft, or possibly

(*b*) That the control spring was not strong enough to ensure at all times the complete travel of the clutch to the ' pawls free ' position. In addition, an error in installation had reduced the distance between the side faces of the pawls and the ratchet ring. It thus seemed possible that an engagement could have been made with the pawls in edge contact, against control spring and sleeve forces, thus resulting in excessive pawl loading.

Notwithstanding the disappointment which accompanied this early failure, its value is now appreciated, for it resulted in a critical re-appraisal of the clutch and clutch controls and of the various operating conditions. Modifications were made, including provision of the device to lock the clutch in the ' pawls free ' position. This lock is remotely actuated and is removed automatically by the initial movement of the gas turbine starting lever.

When trials were resumed, approximately 60 engagements of each clutch were satisfactorily made under different conditions of speed and differential acceleration and finally, 100 engagements were made under the most severe conditions. Records of strain, also differential and absolute speeds, were taken at each engagement and FIG. 35 shows graphs of differential clutch revolutions, speed, acceleration and torque plotted against time, under severe conditions. During an engagement there are two actions, upon passing through synchronism which are completed before the control sleeve can move into the locked position:

(*i*) The pawls must ' pick up ' the ratchet teeth. Hence if synchronism occurs at the instant, just after one pair of pawls has passed a pair

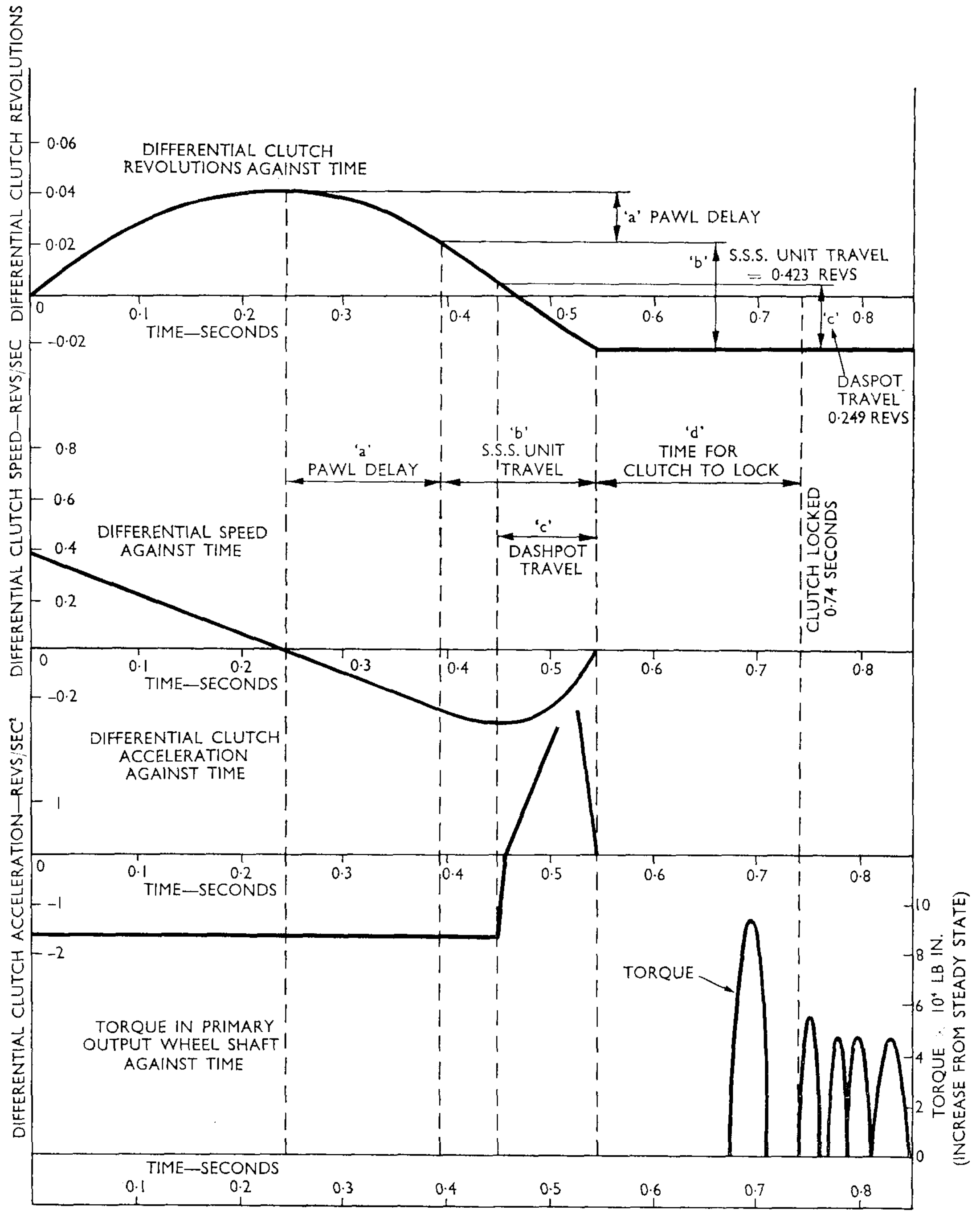

FIG. 35—Y.102—MAIN SYNCHRONIZING CLUTCH—CONDITIONS DURING ENGAGEMENT

of ratchet teeth, the clutch members will rotate relative to each other until the next pair of pawls ' pick up '. This ' pawls delay ' is arbitrary and is between zero and a maximum of 0·0179 revolutions.

(*ii*) The S.S.S. unit moves on the helical splines until the driving teeth engage ; the last part of this movement being against the resistance of the dashpot.

At the end of the engaging travel of the S.S.S. unit, the control sleeve snaps axially to lock the clutch. These positions are shown in FIG. 35 as *a, b, c* and *d*.

From the curve of differential clutch speed against time it is seen that, during this engagement, the speed increased linearly through synchronism until the speed difference between the input and output members was about 0·3 r.p.s and was then decreased by the action of the dashpot. The action of the dashpot is powerful enough to accelerate the slower member and decelerate the faster member, thus bringing into driving contact the teeth of the final reduction gear train just prior to the moment of full engagement of the clutch. If impact should occur at the end of the travel of the S.S.S. unit, the two clutch members will attempt to rebound but this is prevented by the control sleeve which, having a lead on the edge of its locking teeth, as well as backlash, engages and locks the clutch just before the full engagement point is reached. The control sleeve then completes its travel, completely locking the clutch and while this occurs the driving torque increases to its full value.

As two elastic shaft systems are connected, a torsional vibration is excited and it will be seen in FIG. 35 that after the torque stress reached its maximum value following clutch engagement, the driving torque dropped momentarily to zero and rose again : this cycle being repeated with decreasing amplitude, dying away in about six oscillations. Prior to an engagement, the accelerating torque of a gas turbine measured at the strain gauge was found to be negligible and hence these fluctuations in torque can be taken as absolute values. During the return swings of the brief oscillation, gear tooth separation occurs because the oscillating torque is greater than the accelerating torque from the gas turbine at the start of the engagement. The maximum amplitude of the oscillating torque shown in FIG. 35 is $9{\cdot}4 \times 10^4$ lb/in. and this is considerably less than the maximum steady torque with one engine driving.

On completion of these trials the clutch was found to be in completely satisfactory condition. Subsequent experience in the ships has been equally satisfactory.